PRODUCT ENGINEERING

GEORGE F. NORDENHOLT, *Editor*

And There Shall Be Rumors

THE FRENCH HAVE A STORY about a farmer who was in the habit of going to town every Saturday and on his return entertaining his neighbors with the city gossip he had heard. On one of these trips he did not hear any gossip. But he felt it would be an unbearable disappointment to his neighbors if he failed to give them a good story, so as he jogged homeward he conjured a lurid tale.

"A huge fish was caught," he told his eager listeners. "He is so big that his head and his tail lie on the opposite shores of the bay and all the inhabitants of the city are chopping the fish to pieces, filling their baskets with the meat to take home for food."

That evening the farmer noticed many people wending their way cityward. He asked one why so many people were going to town. "Haven't you heard," was the reply. "A huge fish is being chopped to pieces and we are going to get some of it." Not to be outdone, the farmer hastily hitched his horse and wagon and rushed to the city to get his share of the fish.

Such is the history of most gossip. The present continuing crop of rumors relating to national defense and war are largely in this category. Washington is flooded with reporters and correspondents. Each must corral choice tid-bits of gossip or scandal to satisfy editors and the unsatiable appetite of millions of readers. Men who know nothing of science or industry, who do not have even a speaking acquaintance with military and naval problems, "get wind of something," and the story starts the rounds. With each telling it receives further embellishments and superficially sounds more credulous.

Some of these stories are vicious propaganda; many of them are mere gossip; most of them are sheer nonsense. To the credit of the press, few are published. But sooner or later they become bridge table or parlor conversation. The speaker proudly presents his story as being more or less confidential and exclusive. He usually prefixes it with. "I heard that," or "It was told to me by someone who ought to know," or "It must be true because . . ." Enthusiastically, the banker, lawyer, butcher or candlestickmaker "proves" how he knows that the lurid tale must be true.

Now is no time for scandal mongers and gossipers. Now is the time for everyone to put his shoulder to the wheel, and refrain from undermining morale and confidence.

Capital is sometimes made of the evident inexperience of cabinet members or their obvious neglect of their duties. But this is no cause for alarm. It is generally recognized that cabinet appointments usually involve political considerations, and even if the Secretary of the Navy goes off on a political jaunt, the work of preparedness goes on just the same. The real work of preparedness is done not by cabinet secretaries but by men who know their jobs from training and experience, army and navy officers plus a splendid corps of civilian engineers. There are none better.

Chrysler officials and engineers have publicly proclaimed the splendid job done by the engineers of Rockford Arsenal in the design of the 25-ton tank. As Fred Zeder, vice-president in charge of Chrysler Engineering put it, "We don't take second place to any engineering organization, in or out of the automotive industry, but we sure would like to have those engineers at Rockford Arsenal to add to our staff. We never imagined that a 25-ton tank could ride as smoothly and fast as does that 25-ton tank."

In November PRODUCT ENGINEERING we will give details about the fine job that is being done by the Army and Navy engineering personnel and we will introduce to our readers some of the splendid men who are doing the actual work of engineering for national defense.

In the meantime, if you hear a story that derides, criticizes or casts doubt upon our Army or Navy and the mechanization job they are doing, play safe—don't believe it. And in November PRODUCT ENGINEERING we will give you the facts with which to refute such rumors.

Vultee Vanguard 48 Interceptor Pursuit Airplane

STEEL TUBE FUSELAGES

Factors for Economy in Fabrication and Assembly

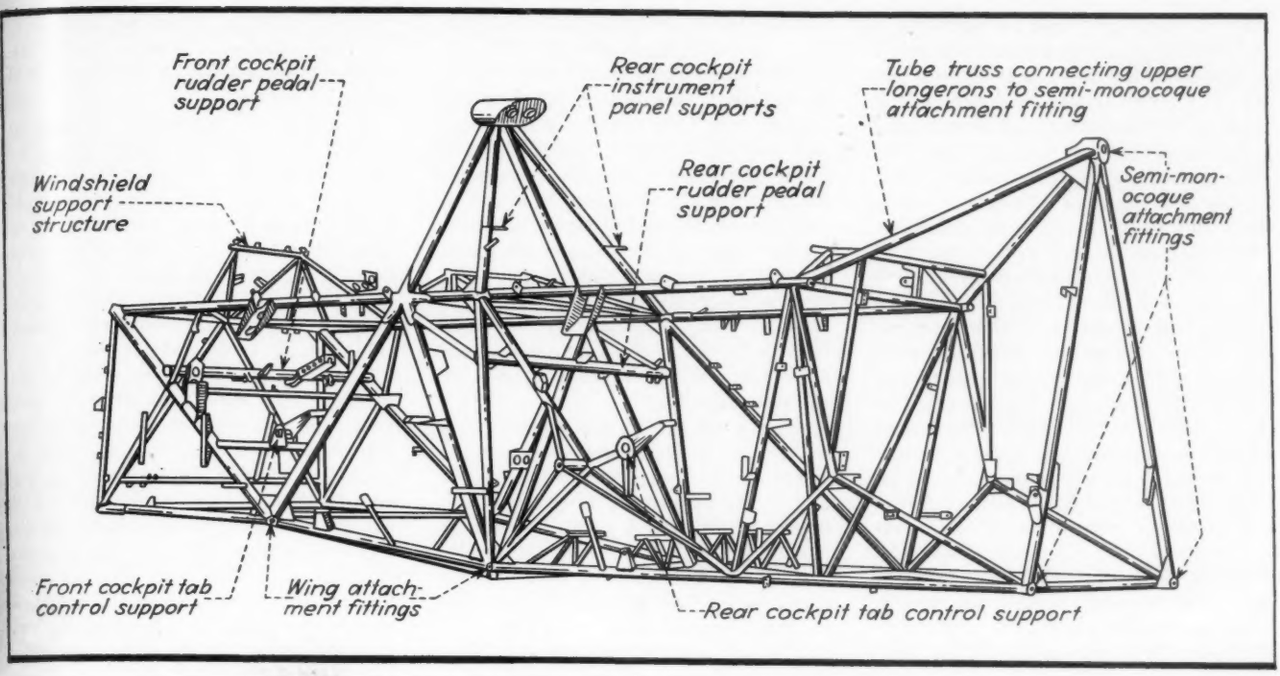
JAMES E. THOMPSON AND T. A. LOWE
Vultee Aircraft, Inc.

PARTICULARLY adapted to the mass production of the smaller types of commercial and military aircraft is the welded steel tube type of fuselage. This form of construction permits fabricating the fuselage, the basic structural element of the airplane, in simple, accessible jigs, and makes it possible to apply the fuselage skin or covering as the last step in the assembly of the airplane. Local concentrations of work in the assembly line are avoided, thus eliminating one of the major problems of mass production, which must be predicated upon a smooth flow of work being applied to the airplane as it passes through the various stages of fabrication and assembly.

The purely abstract considerations involved in the design of welded steel tube structures are identical to those common to other phases of structural engineering. The stress analysis methods applicable to steel tube fuselages are but modifications of the established procedure used for analyzing bridge trusses and similar structures. After the directions and magnitudes of the loads involved are known, and the required margins of safety established, the determination of tube sizes and joint requirements become comparatively simple matters. The tubes can be joined by welding with either the electric arc or gas. The process differs from other applications of welding only in that greater skill is required in working with thin tube gages to maintain required accuracy in alignment.

When laying out a welded steel tube fuselage it is of prime importance that the designer have a practical knowledge of all the factors affecting the economical production of the structure. These factors begin to influence the layout of the basic structure at the time when it is simply a geometrical concept of lines and points, and continue to shape the design throughout to the determination of major attachment fittings, selection of the method of covering the tube structure, and finally of detail fittings and attachment points required for controls and equipment. If these factors are neglected anywhere in the progress of the design, beginning with the geometric layout of the basic structure and concluding with the design of the last attachment fitting, the fundamental economy of welded steel tube structures will be sacrificed.

In the design of a welded steel tube fuselage the first step is the determination of the fuselage form to be used. The basic fuselage structure may consist of two main forms: the all-steel-tube structure wherein the complete primary structure is a welded steel


Front cockpit rudder pedal support
Rear cockpit instrument panel supports
Tube truss connecting upper longerons to semi-monocoque attachment fitting
Windshield support structure
Rear cockpit rudder pedal support
Semi-monocoque attachment fittings
Front cockpit tab control support
Wing attachment fittings
Rear cockpit tab control support

Tube structure for a composite-type fuselage for a two-place, tandem training airplane

Junction of steel-tube structure and semi-monocoque structure in typical composite-type fuselage

tube truss, or the so-called "composite" structure made by joining together a welded steel tube forward portion which includes the cockpits, and an aluminum-alloy semi-monocoque aft structure.

Some designers favor the all-steel-tube design, believing that a single fabrication process throughout for the basic structure facilitates production by simplifying tooling, minimizing the factory equipment required, and eliminating the fittings necessary for attachments of the semi-monocoque section. This form is also desirable because access panels extending the full length of the fuselage are removable thus providing ample room for installation operations during the airplane assembly. The panels can also be removed for inspection and service.

Another group of designers believe that complete access to the extreme aft portion of the fuselage is rarely required. This group prefer the composite type in order to take advantage of its lighter weight, higher rate of production in factories having facilities for sheet metal construction, and the economy gained by eliminating the necessity of fairing side panels and their supports to the aft portion of the fuselage.

Regardless of the type selected, however, there are certain factors involving the basic geometry of the tube structure

layout which must not be neglected if economical production is to be achieved.

Consider the all-steel-tube structure first. Because of the need for cockpit spaces, landing gear and lower wing attachment points, the four longeron type structure is almost always used.

The four longeron tube type of structure inherently provides the designer with a choice of any two of the four sides as major sub-assemblies of both the fore and aft sections. The practice of dividing the structure into fore and aft sections (to be later joined by welding) is universally followed, partly for production breakdown reasons and also to save weight by reducing tube sizes in the rear section. In cabin type airplanes it is also necessary to build an irregularly shaped front section because of passenger compartment space requirements, with the result that the structure is divided aft of the cabin to permit economical weight distribution, and to simplify tooling, as well as to facilitate final alignment of the tube structure.

Naturally, to include two longerons and the tubes connecting same, a choice must be made to build either the top and bottom frames or the side frames, as sub-assemblies. The forward section bottom frame is usually well braced and may be depended upon to hold its shape and resist distortion through the major assembly. The top frame, however, is usually poorly adapted to retaining its shape as a sub-assembly panel, because of the necessity of providing clear unobstructed bays for cockpits in open cockpit airplanes. At the open bays deflection of the frame may occur after removal of the frame from the sub-assembly jig.

After a consideration of these factors it is logical to design the side frames as sub-assemblies, consisting of the upper and lower longerons, inter-connecting tubes and fittings. A further advantage is obtained in that the sides are usually symmetrically braced so that welding distortion and shrinkage are practically the same for both sides when consistent welding procedures are followed. With proper guidance and indexing the side frames, being well braced, may be depended upon to determine accurately the final structure assembly, thus eliminating the need of an elaborate final assembly jig.

The choice of sub-assembly frames to form the aft section, however, may be the top and bottom frames rather than the side frames. The decision depends largely upon the type and location of empennage and tail-wheel attachment fittings. In any event, the rigidity of any side of the aft section is usually sufficient to maintain its shape and permit sub-assembly frame fabrication.

The Pratt truss is shown in all of the fuselage layouts illustrated in this article as this type of truss offers the advantages of normal vertical lines to work from during drafting and tooling layout. However, the Warren truss will usually consist of fewer tubes, and for this reason may be lighter for a given fuselage size. The type of truss used will actually depend upon the required attachment points as governed by factors extraneous to the tube geometry, such as placement of the wing to obtain the correct airplane balance.

Geometry of All-Steel Tube Fuselages

IN THE LINE DRAWINGS of tube layouts in Figs. 1 to 6 are shown logical development of the geometry of an all-steel-tube fuselage design. The sequence of these figures is based upon the design's desirability from the standpoints of fabrication and tooling, beginning with Fig. 1 as the most desirable. All illustrations are for a fuselage suitable for a two-place, tandem, open-cockpit airplane, but the principles discussed are applicable to all types and therefore detail treatment of the cabin-type airplane has been omitted purposely. Note that in all designs the object is to accomplish the same result, namely to retain all frames in true flat planes whenever possible, especially the side frame.

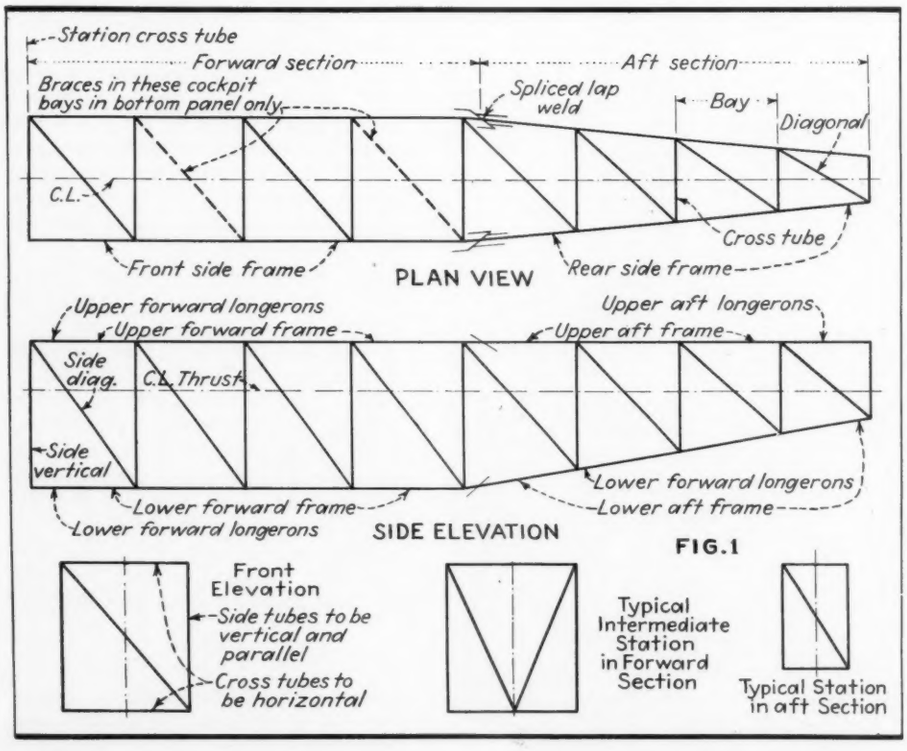


FIG. 1

All-steel-tube structure that is ideally adapted for economical mass production. From the plan view in Fig. 1, it can be seen that both the upper and the lower longerons of the forward section are perfectly straight in all planes, and are also parallel to and equidistant from the centerline of the airplane for the entire length of this section. The upper and the lower longerons of the aft section lie in vertical planes, with the bend in the longerons at a position adjacent to the welded junction of the forward and aft sections. All longerons are straight lines excepting where the bends occur at the junction just forward of the splice. Also note that these bends are at the same station in both the plan view and the side elevation, and because of this fact the necessary tooling and production operations connected with this fuselage can be readily accomplished.

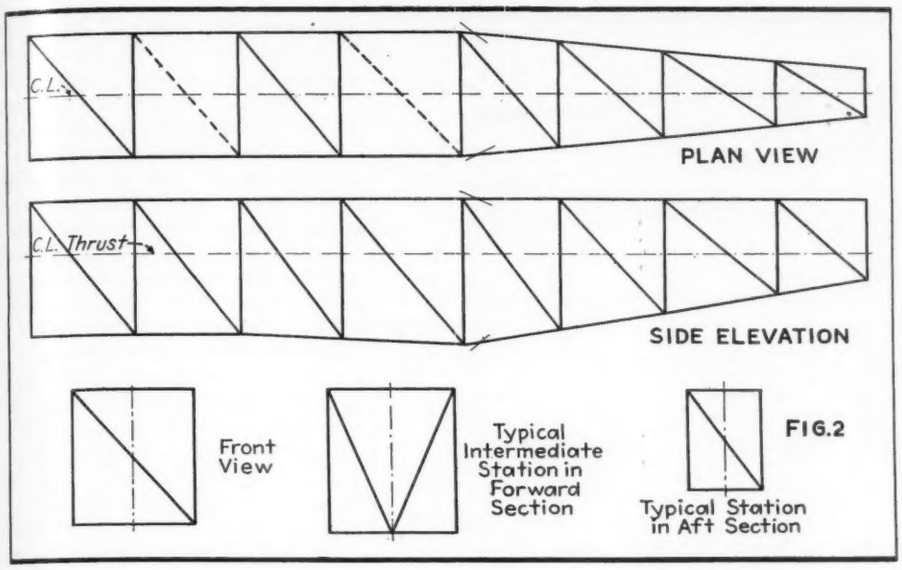


FIG.2

Upper longerons in the side elevation, Fig. 2, are straight and parallel to centerline of thrust. In the lower longerons one bend is at the same station as the bend in the plan view, another bend is at an intermediate station further forward. If possible, one section of the forward lower longerons should be parallel to the centerline of thrust. In the plan view longerons are straight, equidistant from and also parallel to the airplane's centerline. Upper and lower longerons of aft section lie in a vertical plane. All longerons with the exception of the lower forward are straight between junction points. All of the side frames are in one plane.

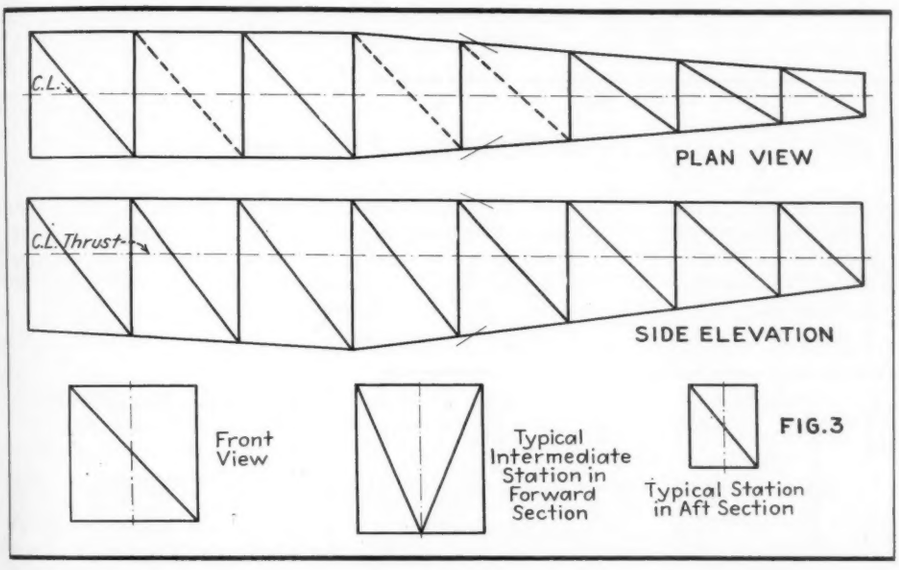


FIG.3

Fuselage in which the upper and the lower longerons are straight in the plan view of the front section, shown in Fig. 3, with the exception of the one bend at the station forward of the junction of the forward and the aft sections. The aft portions of the forward longerons falls on a straight line with the aft section longerons. The front and the rear sections have vertical sides. In the side elevation the upper longerons are perfectly straight for the length of the fuselage. They are also parallel to the centerline of the thrust. One bend is made in the lower longerons, it is located at the same station at which the bend occurs in the plan view.

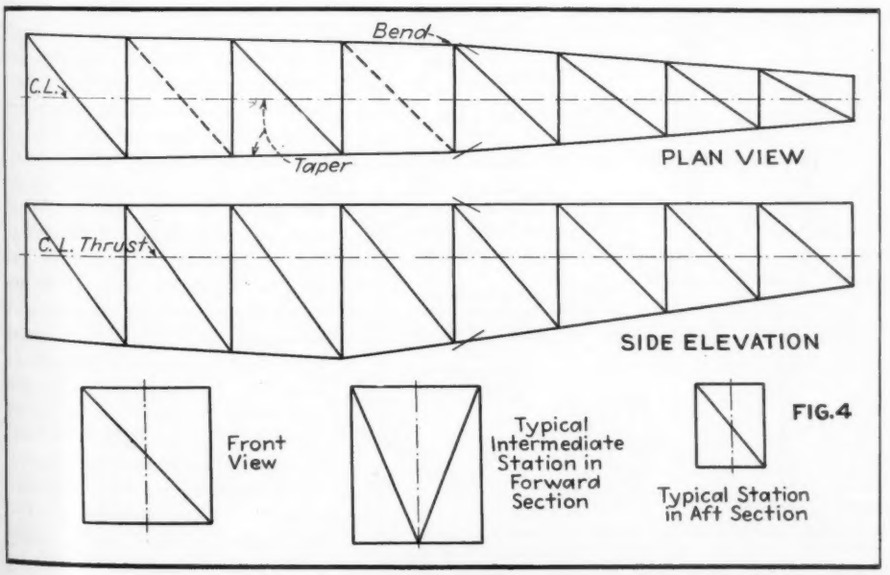


FIG.4

Layout of fuselage similar to that in Fig. 3, but not quite as well adapted to production is shown in Fig. 4. Although sides are vertical, they taper in toward the rear, starting at the front station. The welded junction of fore and aft sections is at the identical station for all longerons. This type of construction introduces some difficulties. Drafting becomes complicated since tubes will be encountered in many layouts that are not in plane of the paper, and must be projected to obtain true views. The same difficulty will be encountered by tool designers when laying out assembly jigs.

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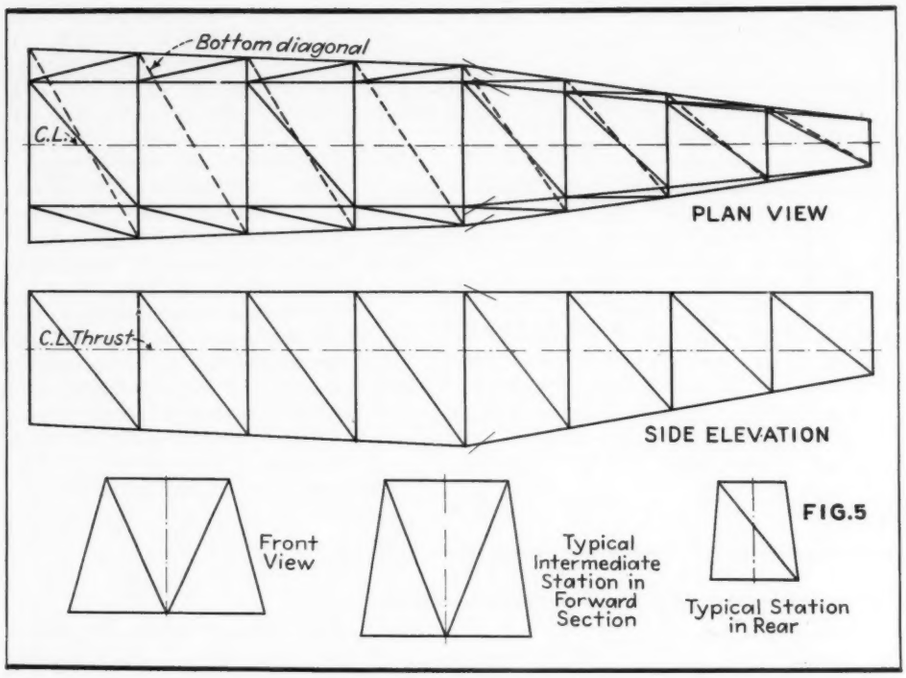


FIG.5

Upper longerons in plan view in the forward section, Fig. 5, are straight and parallel to center line of airplane; lower longerons are also straight but not parallel with upper ones, being tapered in toward the aft end so that side frames are in a "twisted" plane that complicates all layout and tooling work. The same condition exists with regard to longerons in the aft section. In side view upper longerons are perfectly straight, and parallel to line of thrust and thus present the only truly unbroken plane in the fuselage, but unfortunately one that cannot be built up as a sub-assembly because of the unbraced cockpit openings, precluding the necessary rigidity. A design of this nature should be avoided because of the high tooling and production expense involved.

Geometry of Composite-Type Tubular Fuselages

COMPOSITE-TYPE FUSELAGES shown in Figs. 6 and 7, are advanced developments wherein the steel-tube structure terminates immediately aft of the rear cockpit, and the aft portion of the fuselage is formed by an aluminum-alloy semi-monocoque structure. A composite type fuselage may employ any one of the basic steel tube layouts shown in Figs. 1 to 5, although it is quite obvious that it is still important to use a simple tube structure. The major additional consideration lies in the method of attachment used between steel tube structure and semi-monocoque structure.

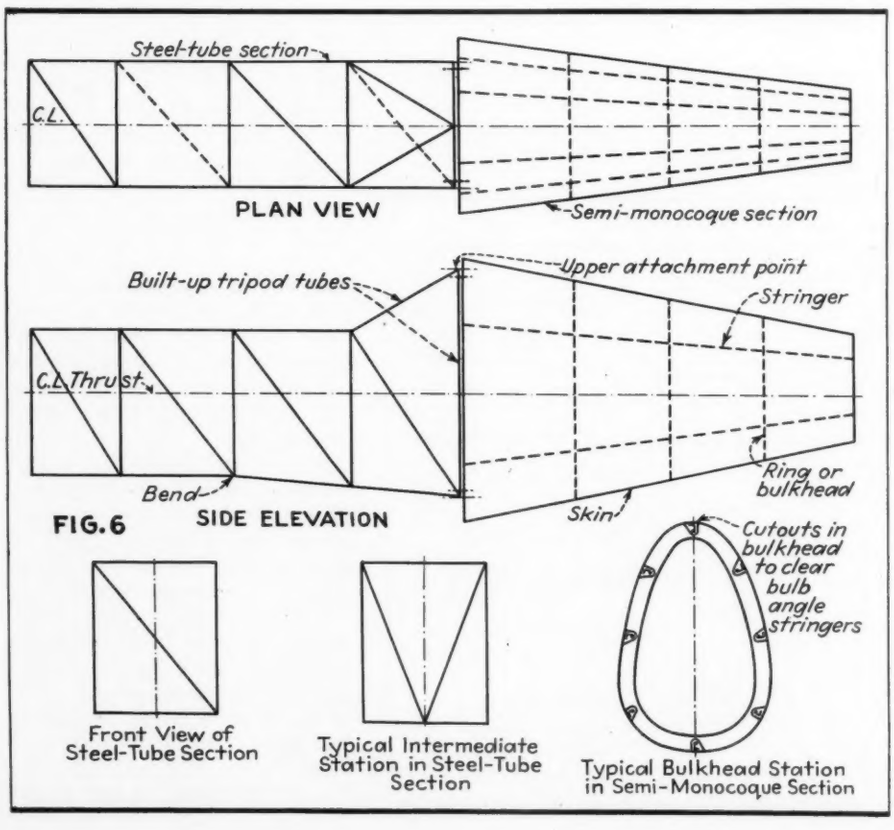


FIG.6

Three-point attachment design for connecting a steel tube truss structure to a semi-monocoque section similar to the type that is illustrated in Fig. 6 is considered by airplane designers to be the most practical, since this method of construction permits placing the attachment fittings in favorable position such that much of the torsion and the bending resistance which is inherent in the semi-monocoque type of structure can be utilized to good advantage, which is to say, that these attachment points are placed as wide apart as possible in the structure thus obtaining a particularly favorable result, the achievement of structural strength with good weight economy. The benefits derived from this three-point method of attaching a semi-monocoque structure, however, are offset somewhat by a slight disadvantage in that a separate sub-assembly is required for the triangular tube truss which connects the longerons to the upper semi-monocoque fitting.

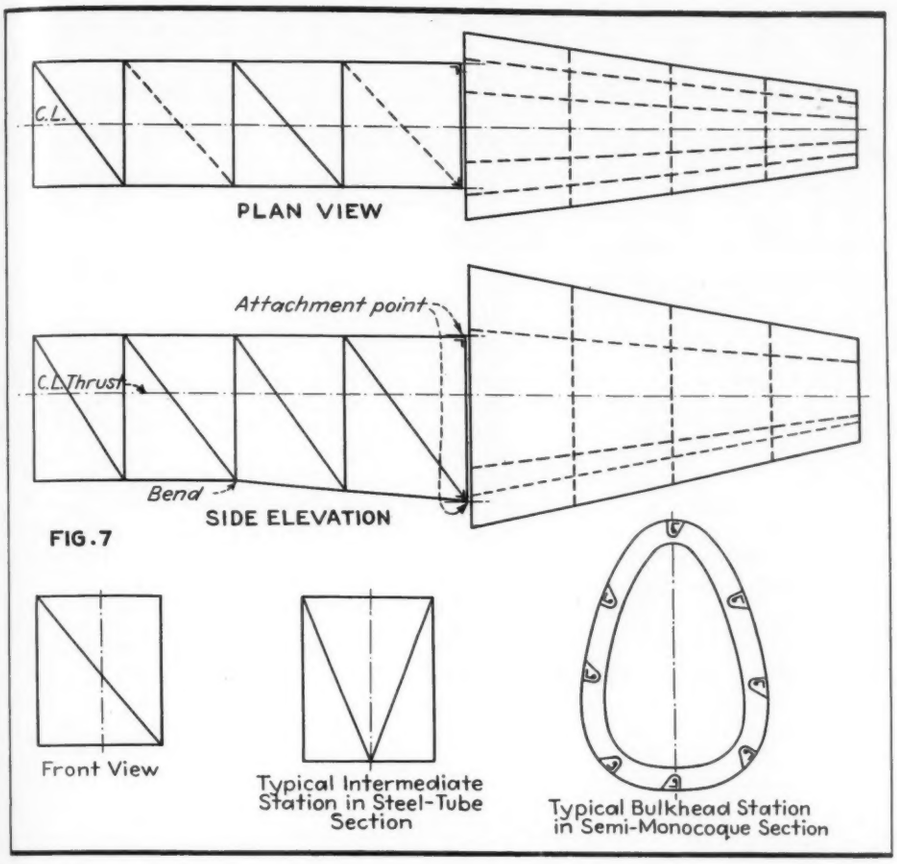
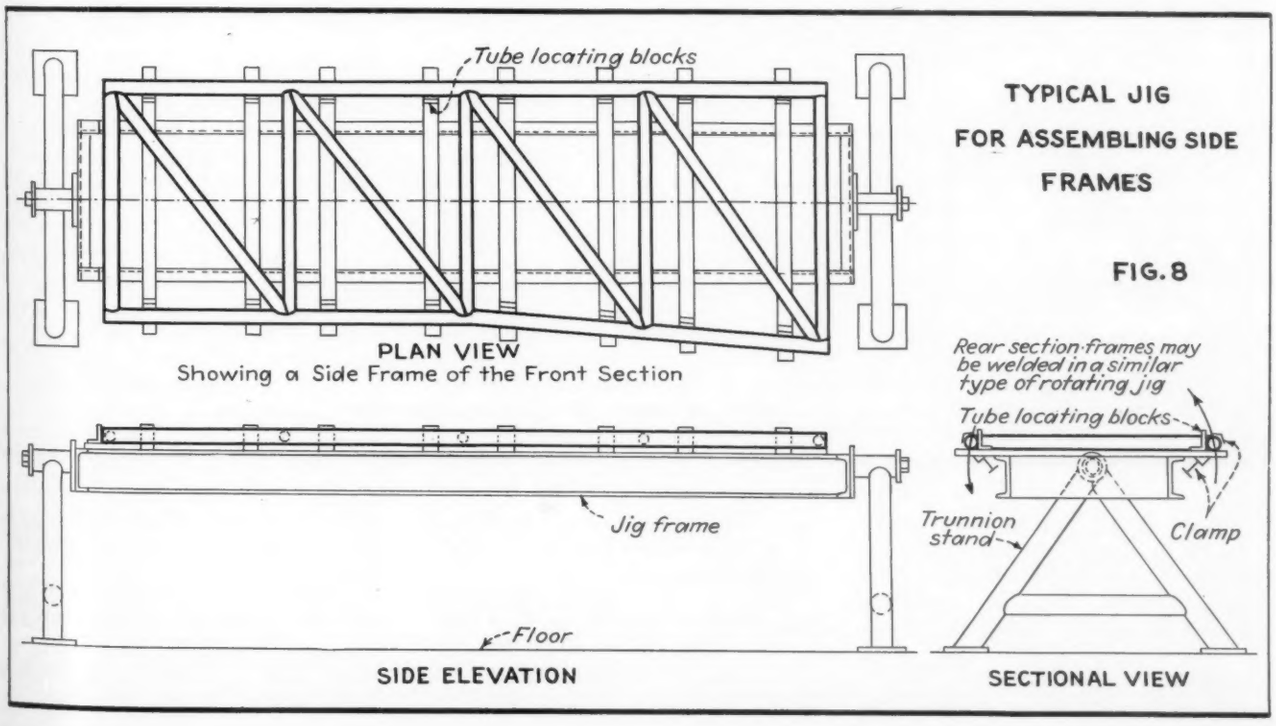


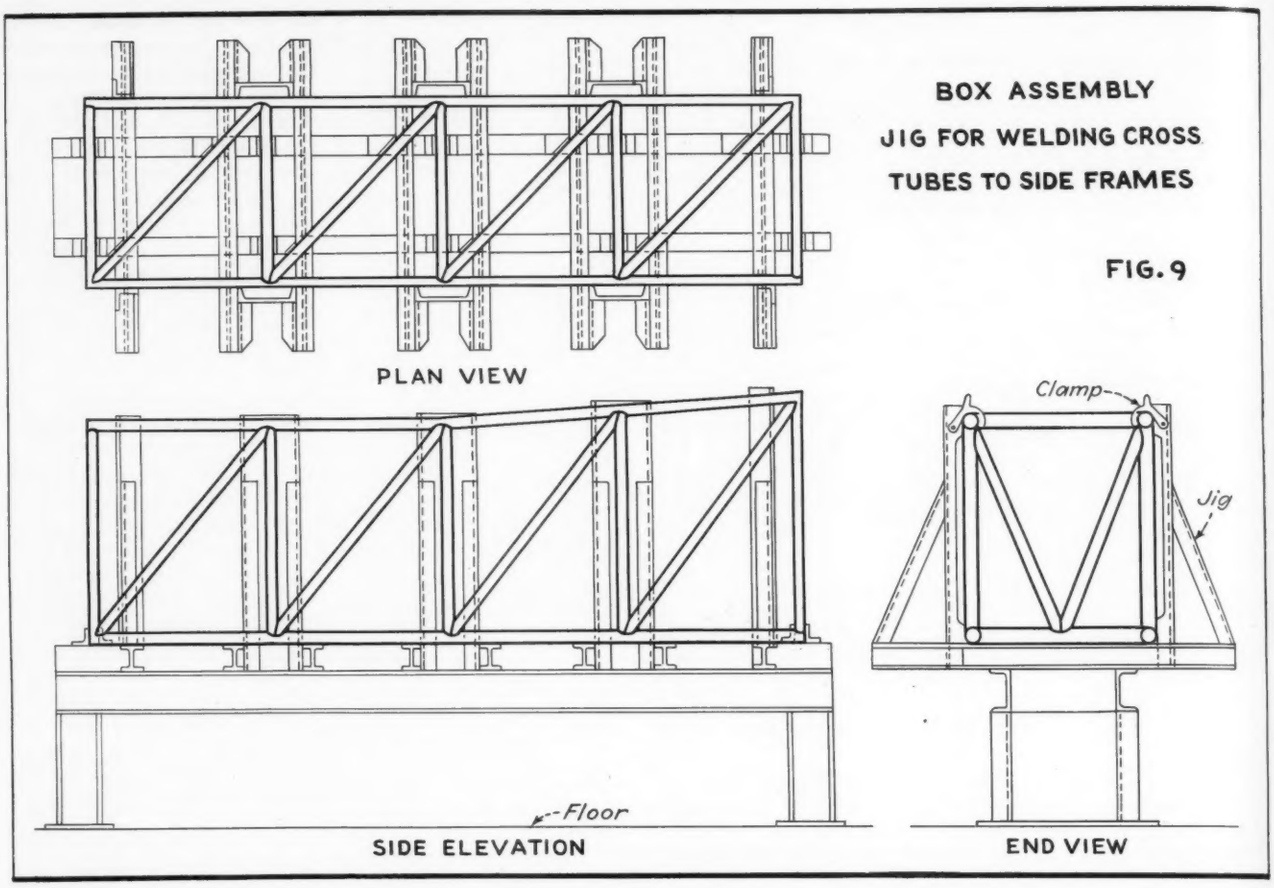
FIG. 7

Alternative method of semi-monocoque attachment shown in Fig. 7, involving four attachment fittings formed by terminations of longerons, avoids necessity of a separate tube sub-assembly for upper fitting. However, this may not be as satisfactory from a structural standpoint, as the material of the semi-monocoque section is not always as evenly loaded as with three-point attachment, and it would probably be necessary to use a slightly heavier gage sheet for the skin covering. To comprehend this factor, it should be understood that the semi-monocoque structure is rarely, if ever, a round section, but rather a sort of flattened elliptical section having a smaller minor axis at its upper end to meet the requirements of aerodynamic form, particularly with the smaller two-place tandem type of airplane wherein the fuselage must be deep and narrow. If the semi-monocoque structure could be designed as a true circular section, then the four point attachment would have definite structural advantages.

Basic Forms of Assembly Jigs for Tubular Fuselages

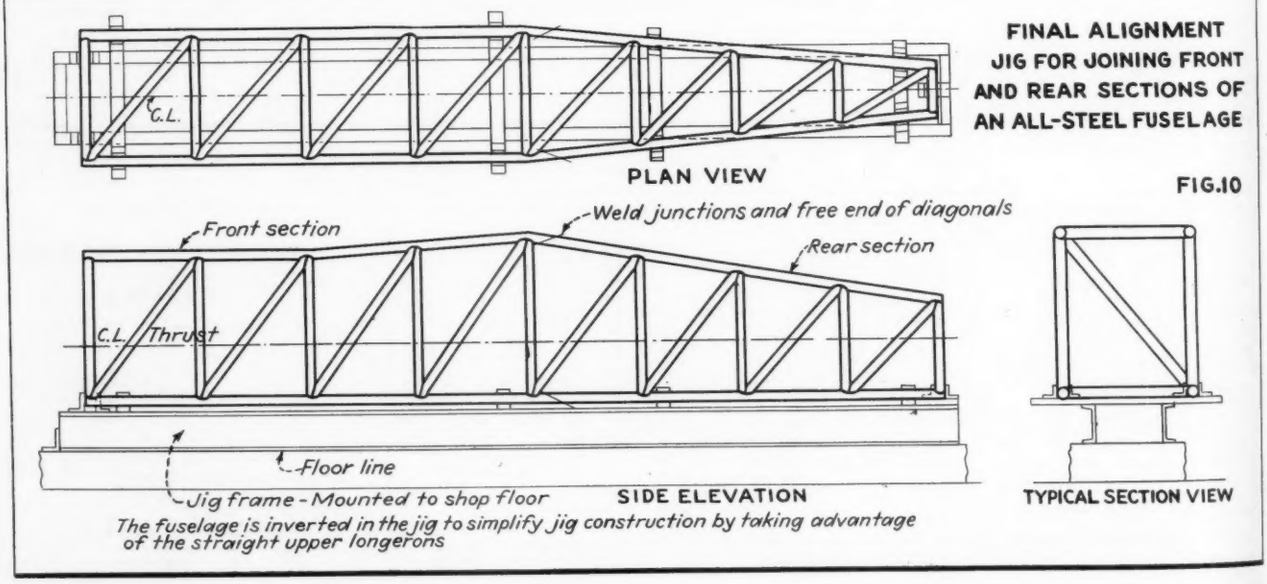
ASSEMBLY JIGS shown in Figs. 8, 9 and 10 illustrate basic forms suitable for steel tube fuselage fabrication, and serve to provide practical illustration of many of the points which have been discussed. *(Continued on next page)*





BOX ASSEMBLY JIG FOR WELDING CROSS TUBES TO SIDE FRAMES

FIG. 9


C.L.
PLAN VIEW
Front section
Weld junctions and free end of diagonals
Rear section
C.L. Thrust
Floor line
Jig frame - Mounted to shop floor
SIDE ELEVATION
TYPICAL SECTION VIEW
The fuselage is inverted in the jig to simplify jig construction by taking advantage of the straight upper longerons

FINAL ALIGNMENT JIG FOR JOINING FRONT AND REAR SECTIONS OF AN ALL-STEEL FUSELAGE

FIG. 10

The designer's next problem, following the completion of the basic tube structure layout, is the selection of the required major fittings for attachment of principal units to the fuselage. These are: engine mount attachment; wing attachment, semi-monocoque attachment, in the case of a composite fuselage; overturning structure, in the case of an open-cockpit airplane; and empennage attachment.

Many of the important production design factors affecting each of these, together with the factors influencing engine mount structures, will be discussed in the next installment of this series of articles, which will appear in the November number of PRODUCT ENGINEERING.

Subsequent installments will deal with the design of minor fittings and equipment installation provisions; fuselage covering methods; and design considerations relating to tooling.

PHILOSOPHY OF DESIGN

The Foundation for Better Planning

C. M. ASHLEY

Product Development Engineer, Carrier Corporation

SUCCESS of an industrial research and development organization depends not only upon the ability and intelligence of its members and on the effectiveness of their organization. Equally important is a proper working philosophy as the guide to action for each individual and for the group as a whole. The development engineer must deal with nature in a most imperfect state where theory is frequently hidden from sight by the complexity of interactions involved. Problems with which he wrestles are usually too complex for exact analysis. Experience teaches the development man things which his intellect would never have told him. He learns the proper balance between daring and caution, idealism and realism, theory and practice. His way of thinking must be pragmatic in character, based upon an evaluation of risks and the probabilities of a given situation.

This Thing Called Perfection

The perfectionist is never happy as a development engineer. He quickly learns that perfection in any detail is difficult to obtain and requires an inordinate amount of time. He is soon forced to one of two alternatives. Either he perseveres, seeking perfection, in which case he ceases after a time to be a useful development engineer, or else he gives up his ideal of perfection, in which case he is no longer a perfectionist. At the other extreme is the man who is easily satisfied with an imperfect achievement and never realizes the possibilities of the thing which he is developing. Less complacent competition sweeps past this man.

Perhaps the ideal attitude is that of the deferred perfectionist. While never wholly satisfied with present forms, he realizes perfection is an objective which cannot completely be attained and is content to strive for worth-while improvements. The problem of deciding when a development has reached a practical stopping point requires fine judgment. The design must be viewed objectively from the point of view of both the user and the salesman. Two questions must be asked: First, is the equipment developed commercially practical? Second, have possibilities of trouble been reasonably eliminated?

One reason why perfection cannot be taken too seriously is that it doesn't stay put. At best it is a mental image. Too often the engineer finds that his mental image has no reflection in reality, that a perfectly conceived and executed design leaves the field cold when some purely practical objection, such as cost or lack of market, acts as an insuperable bar to acceptance. Another quality of perfection is its growth. As progress is made our imagination reaches out still further ahead. Our ideas of perfection in equipment are constantly being tempered and conditioned by changing outside influences such as customer reaction, practical workability, or competition.

So the wise development man recognizes perfection as that something within himself which makes him dissatisfied with present forms, rather than an absolute objective. He learns to control this dissatisfaction to yield maximum returns for the time and monev with which he has to work.

When Trouble Comes

Since perfection is never achieved, the development engineer is constantly beset by troubles. Equipment design is a strange mixture of theory and practice and often the theory comes only as the rationalization of recognized practice. In many cases it is only by repeated trials of a new design element that success can ultimately be won out of failure. Even the minor modification of a design in a thoroughly explored field may be beset by trouble. Consider the new model of an automobile, as an example. Let the motor manufacturer change anything more fundamental than the decoration on the hood and he brings down on himself a host of "weaknesses" despite endless trials in the laboratory, proving ground and on the road. If this can happen, how much greater is the danger of trouble in a field not nearly so well stabilized nor able to give a new design adequate advance trial? What chance of being free from troubles has a new design, even when relatively simple, when put out with no field trials at all?

Trouble has another characteristic. No matter how successful 999 of 1,000 elements of a machine are, if the one remaining part gives trouble that is all that matters. A man may have a car whose motor functions perfectly but he is still thoroughly annoyed by a weak ignition system, a shorted starter, or a grabbing clutch. Yet the "bugs" in that car may amount to only one-hundredth of one per cent of the design problems which have been solved successfully.

What attitude must be taken? First, trouble must be expected from any change, however small. Second, the likelihood of trouble increases with the magnitude and with the novelty of the change. Third, the chances of catching the trouble in the laboratory stage depend upon the extent and intelligence of the proof tests. Fourth, no amount of proof-testing will catch all of the troubles. Fifth, the wise general always lays his plans for a retreat before he starts his advance.

When a change is made, ask three questions: One, what are the chances of trouble? Two, how can the trouble be discovered? Three, what alternatives are there in case trouble develops? Regarding the third, perhaps there are other methods for achieving the same result that do not seem quite as cleanly designed but which may be less novel, thus safer. Again, the changed part may be sub-divided so that replacement is less expensive. Or particularly easy access might be arranged to cut the cost of a field repair.

The role of the development engineer becomes very much like that of the actuary who tries to gage the magnitude of the risk and sets premiums accordingly. Where the production of a machine part is small, it may not be expensive to take large risks. Where the risk possibility is great, the production must not be permitted to grow too large before extensive tests have proved the new design and reduced the risk to a minimum.

One of the most difficult psychological problems is for a man accustomed to designing for small quantity production, to shift to designing for mass production. Where before he had the utmost freedom in making changes immediately as experience dictated, now he must stop to consider questions of cost, inter-

changeability, field replacement, confusion in manufacturing and marketing, and a host of others.

There is one other aspect of trouble that certainly deserves mention. It seldom or never occurs where it is expected. This is popularly recognized and is not so difficult to understand when it is remembered that anticipated trouble is usually guarded against. It is the unanticipated trouble against which we cannot protect ourselves. Still, we know from experience that we must expect it and prepare to meet it from whatever quarter it may come—by keeping designs as flexible and adaptable as is in any way consistent with the factors of cost and results.

The Time Factor

Pure research does not generally recognize time as one of its limitations. Development, on the other hand, meets constant pressure from management to conform its projects to rigid time schedules. Frequently these schedules are dictated not by the time required to carry on the development in an orderly manner, but by the exigencies of the commercial situation, such as competition, production demands, sales needs. It is hard for people outside of the development group to understand why schedules cannot be set up and met, and it is equally hard for those within to set them up and meet them.

Any sort of schedule assumes the success of each step and of the development as a whole, for failure would inevitably retard the work. Actually, however, failure is the customary initial reward of development. In fact, design engineers meet with so many failures in this line of business that they have to be optimists by nature. Otherwise they wouldn't have the courage to keep plugging. Success comes as a rule out of a series of time-consuming failures.

Most developments must fulfill not only one objective, but dozens simultaneously, such as cost, power demand, weight, space occupied, accessibility, quietness, good appearance, and reliability. The failure to meet any one of these may result in failure of the design as a whole. After extensive study it may be found that the objectives are unattainable. They must then be restated and the work started all over again. It is not to be wondered, therefore, that development problems, even those along conventional lines, should be hard to schedule. As for the radically new things, schedules had best be forgotten until some success becomes assured.

Every orderly time estimate of a development is based upon the summation of the time estimates for each separate step. You may allow for failures by figuring on one, three or five trials to reach a successful solution of a given step, but there is no assurance whatever that this will be the number actually required.

It is commonplace to observe how simple an accomplished result seems compared with the labor and circumlocutions by which it was reached. Experience seems to indicate that developments take about three times as long as there seems any good reason that they should take. But it does not do to set up a schedule which allows this extra time for each step, since then the development engineer unconsciously plans a more exhaustive study which uses up more than the time allowed. For the development as a whole this becomes cumulative. The best plan is to set up each step, giving it a reasonable allowance above the requirements visualized at the start of the development. Then, for the development as a whole, an extra time allowance as taught from experience should be made to cover the remaining contingencies.

Savings in time required to consummate a development can be made through more intelligent planning and execution, that is, better organization. This should be the constant aim of a development staff. Too often, however, the development engineer or director is swayed by commercial exigencies into feeling that in a particular case he can "beat the game." He is customarily disappointed. Even short-circuiting the usual procedure seldom helps him in the long run. Although he may be able to complete a development, he is later forced by unfavorable developments from the field to return to his unfinished task.

Time for design is one of the most important factors in the success of an industrial organization. There is no substitute for it. Both management and design engineers must give it due consideration in laying plans.

Development Planning

Planning is the lubricant that keeps a development organization running smoothly. Where proper plans are not made, the essential ingredient, time, is out of phase and the result is more likely to be chaos than accomplishment. The planning must be of three types. First, each development project must be planned so as to arrive at a satisfactory state of conclusion at the desired time. This involves the marshalling of resources and men and working out the time required. In general, all developments follow the same pattern, with variations caused by the magnitude of the design problems and production.

Next comes the planning of the whole development program. Here, all of the projects compete to some extent for the services of the shop, draftsmen and engineers. In this program must be found some way to relieve the "bottlenecks" wherever they occur.

Finally, there is the problem of the long term planning of the product engineering of the company. No company would think of operating without some sort of a budget, even though it knew that the budget would not be followed in all details. In the budget would be an account for the depreciation of all of the physical assets of the company: the machinery, the inventory and the plant. Yet the designs of the products which it makes are seldom considered in depreciation. Some designs may have a useful life of ten or twenty years, but it is normal to expect a life of not more than one to three years before minor changes must be made, and no more than five before something fairly radical is required. In many lines the customer expects, and gets, a new design every year, even though the changes are mostly in the external appearance and gadgets, and the "revolutionary advances" come from the publicity department. It usually takes the better part of a year from the time a minor redesign is approved until it is rolling down the production line, and for a major redesign it may be one to five years, or more. For this reason, it is not enough to wait until some competitor brings out a new design and then try to match or better it.

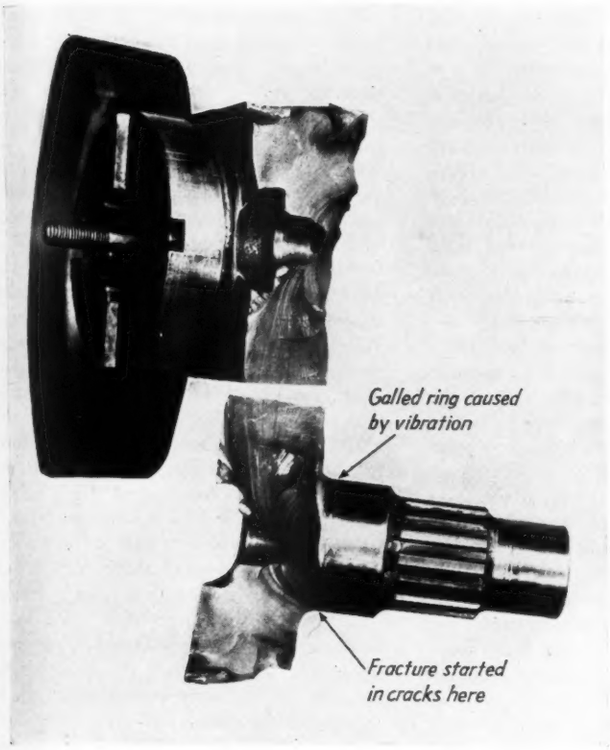
What this adds up to is that a company, in order to stay in the running, must constantly apply fresh study and effort to each of its lines and must be ready to bring out of the laboratory, at the right moment, new models to keep abreast of competitive progress. For success in this, more time than seems necessary will be taken, troubles will bring their headaches, and the work probably won't be perfect when finished. It is the development man's unique privilege to expect, understand and admit these things. And development work can be soundly organized only by men who base their judgment and planning on the underlying abstract factors.

BUSINESS CAN ADJUST its plans and operations to conform to almost any definite system of practices and set of regulations but it cannot plan and carry on under rules which are subject to momentary change without notice or without having any means of even scientifically guessing what the rules may be.

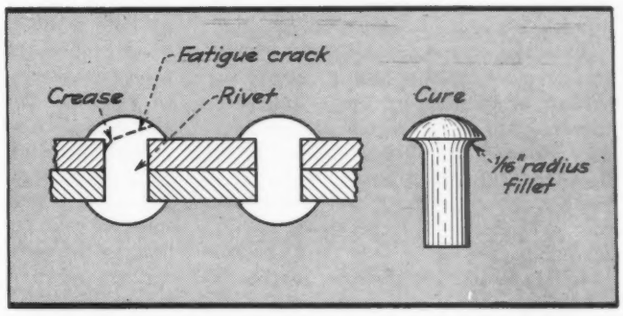
—Raoul E. Desvernine

CAUSES AND CURES

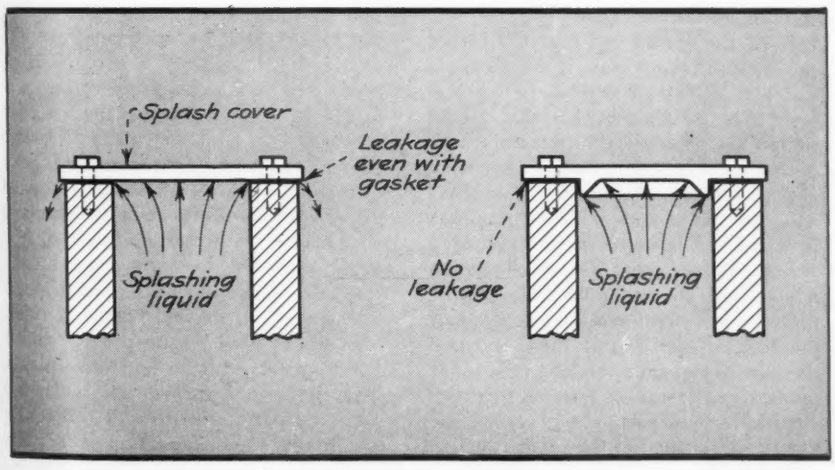
PRODUCT ENGINEERING will pay $3 for each example published in Causes and Cures. Where illustrations are necessary, include drawings, rough sketches or photographs



Torsional fatigue failure of an engine crankshaft started in a narrow galled band around the rear end of the rear pilot and then progressed through the wall of the shaft and finally to the end of the shaft. After close study, failure was not attributed to inherently faulty material nor to the method of quenching in heat treatment. Chief cause of the failure was considered to be the galled ring, in which two cracks originated. The ring appears to have been caused by repeated impacts, such as vibration of the end of the front shaft, which embrittled the surface metal.



Riveted joints subject to vibratory or repeated loads have been found to fail by fatigue when the rivet is driven in an oversize hole, causing a crease under the head of the rivet as shown in the diagram. Failure starts in crease, works through head. The remedy is to design the rivet with a $\frac{1}{16}$-in. fillet.



Leakage of liquids through joints under the splash cover of a container, even when the cover has a tight fitting gasket, is a common occurrence when the liquid in the container splashes violently. An easy remedy for this trouble has been found by designing the splash cover with a projecting lip around the inside. This lip makes a loose male fit with the opening of the container. The splashing liquid then strikes this lip and falls back into the container without striking the gasket with enough force to leak out.

Spring-loaded clock timers, operated in humid, high-temperature conditions, were found by a package machinery manufacturer to be inaccurate. These conditions caused variations in spring characteristics. Cure was to substitute compensated bimetallic timer.

Steam boiler accidents investigated in the past showed that employees were often seriously scalded while in the act of shutting off the water glass valves following failures of the gage glasses. No doubt the seriousness of these injuries was contributed to by the fact that the valve stems had fine threads which required several turns of the handwheel to close the valves. To correct this, gage-glass valves that can be closed with less than one turn of the handle were substituted. Consideration should be given to the use of such quick closing valves, especially on high pressure steam equipment.

MOTOR NOISE

As Influenced by the Design of Rotor

CARL E. JOHNSON
President, Sterling Electric Motors, Inc.

FOR many years designers of induction motors paid little attention to noise; reliability, cost and performance characteristics were stressed. As the public became more "noise conscious" manufacturers were forced to devote more attention to the possibilities of reducing or eliminating noises inherent in their products. Noises in induction motors come from mechanical, windage and magnetic sources, and some of these noises can, when prevalent, be very objectionable.

Induction motor noises from mechanical sources usually are not great in volume or of high frequency and are, therefore, tolerable. Good design, perfect balance of rotating parts, good manufacturing practice and the selection of proper bearings well mounted, all serve to keep these noises to a minimum.

Windage, often a common source of noise, is caused mainly by vibrating air columns produced by fans and the passage of this air through the ventilating ducts. This noise can be, and usually is, kept at a minimum by proper design of the air passages, the number and shape of blades of fans with respect to the passages and obstructions so as to eliminate as far as possible what are known as "siren effects." Windage noise is easily distinguished from other noises, being the predominating noise when the motor is coasting.

Magnetic noises, usually present in varying degrees, are generally of frequencies above 1,000 cycles per sec. and, therefore, are more audible and annoying. They originate from a number of causes such as stator vibration arising from flux distribution, tooth frequency by reason of the number of slots in the stator and the number of teeth in the rotor, and also the magnetic density and air gaps. The mechanical stiffness and shape of the teeth in both the rotor and the stator lamination will also have an effect.

From this brief analysis of the noise problem it is apparent that results at least cost can best be effected by eliminating as many as possible of the more audible and disturbing noises, most of which can be traced to the action between rotor and stator slots and teeth.

All of the rotors in early designs of induction motors were made with what are known as straight bars. It was found that the magnetic action between the straight rotor bars and straight stator teeth resulted in the "abrupt breaking" off of the flux, which produced tooth vibration and thereby objectionable high pitched noises. After some analysis it was determined that, if the rotor bars or stator slots were "skewed" a small amount, the magnetic flux would be "sheared off" instead of broken abruptly as the rotor revolved, thereby reducing the excessive vibration of the teeth of both stator and rotor. It was also found that a greater angle of skew of the bar gave more shearing effect and a further reduction in noise but that, if the bars were skewed too much, the characteristics of the motor would suffer. Designers were then compelled to select an acceptable compromise between noise and performance. The answer was that, performance usually being the more important, a tolerable degree of noise had to be accepted. The straight skewed construction being the best solution at the time was generally adopted, and so it remained until designers became more "noise conscious" and a better solution could be found.

With the knowledge that the larger skew would reduce noise it was a simple and logical step to work out a herringbone type of construction that would permit a greater skew of the rotor bars with the desired greater "shearing effect" without affecting the characteristics of the motor. This herringbone idea offered latitude in the use of a variety of forms but for simplification a standard was adopted that skewed the bars at twice the angle for one-half the axial distance and then returned the bars to a point opposite to the point of starting. Results of tests on this construction confirmed the herringbone theory of reducing the high pitched magnetic noises.

In testing, the objective was to obtain a direct noise performance comparison between straight, 12-deg. standard skewed, and 24-deg. herringbone rotor constructions, shown in Fig. 1, under exactly similar conditions. All three types of rotors were made with standard 57 slot cores, the same size of bar, end rings, diameter, and air gaps—the "bar skew" being the only difference. Otherwise all were interchangeable.

A standard "open type" sleeve bearing, 48 slot stator was wound with a standard 6-pole, 7½-hp. winding. The "open type" was selected in order that a maximum of whatever noise generated would be recorded. Also the 48 slot, 6-pole, irregular grouped, winding was selected because from the magnetic standpoint it is one of the most difficult and troublesome of standard combinations and would therefore fully test the effectiveness of rotor skew on noise generation.

In order to reduce complications and to obtain a more accurate comparison of the effects of rotor skew the cooling fans were left off in all three rotors. All three rotors were tested in the 6-pole stator at no load on 3-phase, 220 volts, 60 cycle current supplied from a large laboratory alternator having good sine wave characteristics. Readings were taken at full load after the motor had attained operating temperature.

All measurements of the noise intensity produced by the motors were made with a General Radio, type 759-A, sound level meter. Sound intensity was taken at five equidistant points from the motors and the arithmetic average of these five readings were taken as being representative of the noise produced. In accordance with the tentative standards of the A.S.A. and A.I.E.E., the microphone was maintained at a distance of one foot from the nearest major surface of the motor. Readings were taken with the response of the meter adjusted to correspond to the response of the human ear at a sound intensity of 40 decibels above a reference level of 10^{-16} watts per sq. cm. at 1,000 c.p.s., as it was felt that this adjustment would furnish an indication most nearly in accord with the sensation produced by the motor in service.

The straight bar rotor produced the greatest observed noise level, or 71.2 decibels as indicated on the meter, a reasonable figure for a 6-pole, 3-phase motor with 48 slot grouping. Taking the 71.2 db sound level as a reference level, the relative sound intensities for the three designs of rotors are as fol-

lows: Straight bar rotor 0.0; 12-deg. skewed bar —11.7; 24-deg. herringbone —15.7 decibels.

These results clearly indicate the advantage of herringbone construction from the point of view of magnetic noise reduction. In the propagation of sound waves, of course, a transfer of energy is involved. The average rate of flow of this energy per unit area normal to the direction of propagation can be conveniently expressed in terms of intensity units, but since the difference in decibels of two sound waves having acoustic intensities I_1 and I_2 is

$$\frac{\text{db}}{10} = \log_{10} (I_2/I_1)$$

it follows that the noise produced by the 24-deg. herringbone rotor, being 4 db below that produced by the 12-deg. skew rotor, represents a power ratio of 2.5, or one motor with a 12-deg. skew rotor releases as much sound power as do two and one-half motors with 24-deg. herringbone rotors.

With the aid of the G.R. 760-A sound analyser noise spectrums were made of all three rotors and the motor noises were separated into their individual frequency components. This analysis is shown graphically in Fig. 1. In these charts the intensity of a given component is shown with reference to the intensity of the strongest component. Thus with the straight rotor the strongest component, Fig. 1(A), was at 1,280 c.p.s., and for purposes of comparison this component is taken as representing zero level while other components are shown as a certain number of db below this reference.

It will be observed from the charts that the noises produced by the motor consist of two distinct groups of frequencies. The first is a lower frequency group corresponding to rotational frequency of the rotor and harmonics of this frequency together with components of frequency equal to that of applied voltage. The second group is of high frequency produced by the rotor and stator teeth and some windage caused by rotor bars passing stator winding openings. The effect of increasing the skew of the rotor bars or shearing the flux is shown by the sound analysis, the intensity of the high frequency components, which are the most objectionable, being considerably reduced.

Another observation during tests that suggested a further advantage of herringbone rotor construction was its tendency not to oscillate as much under sudden changes of load. One-way skewed rotors have more of a lateral "screw effect," tending to screw out of the stator with sudden load changes, a characteristic not noticeable in herringbone construction.

These test results clearly indicate that herringbone rotor construction is a step forward in the art of making quieter induction motors, especially when it is considered that prior to this development the procedure generally followed to obtain even a slight noise reduction in special "silent type" motors was costly, consisting of lowering magnetic densities, increasing air gaps, and providing larger number of slots and coils, which in most designs required larger frames.

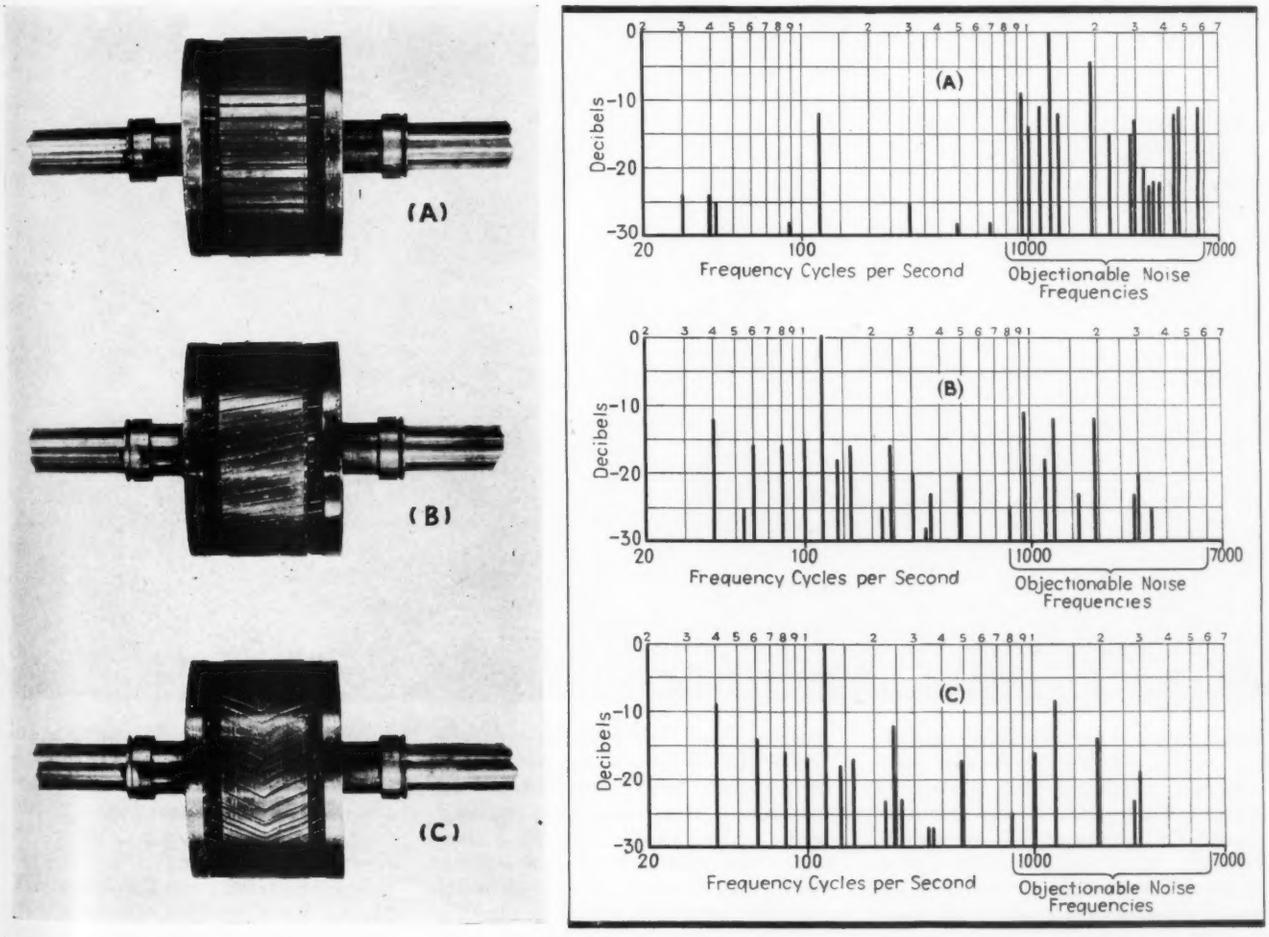
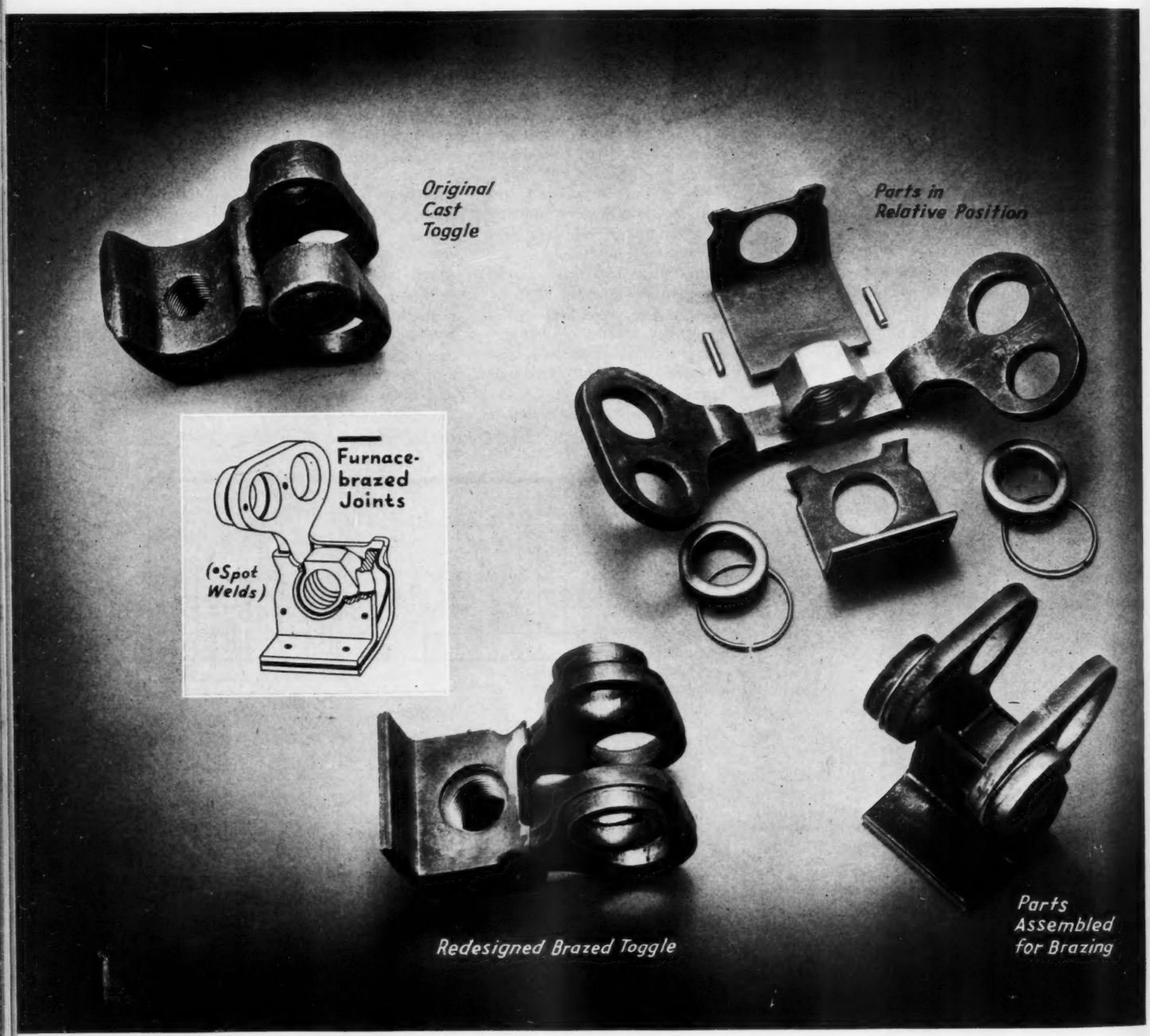


Fig. 1—Types of rotors tested and noise spectrums for motor with each of the three different types, all of which were tested in a 6-pole stator. (A) straight bar rotor, (B) 12 deg. standard skewed bar rotor, (C) 24-deg. herringbone rotor

REDESIGNING FOR FU

H. M. WEBBER, *General Electric Company*, L. SWA



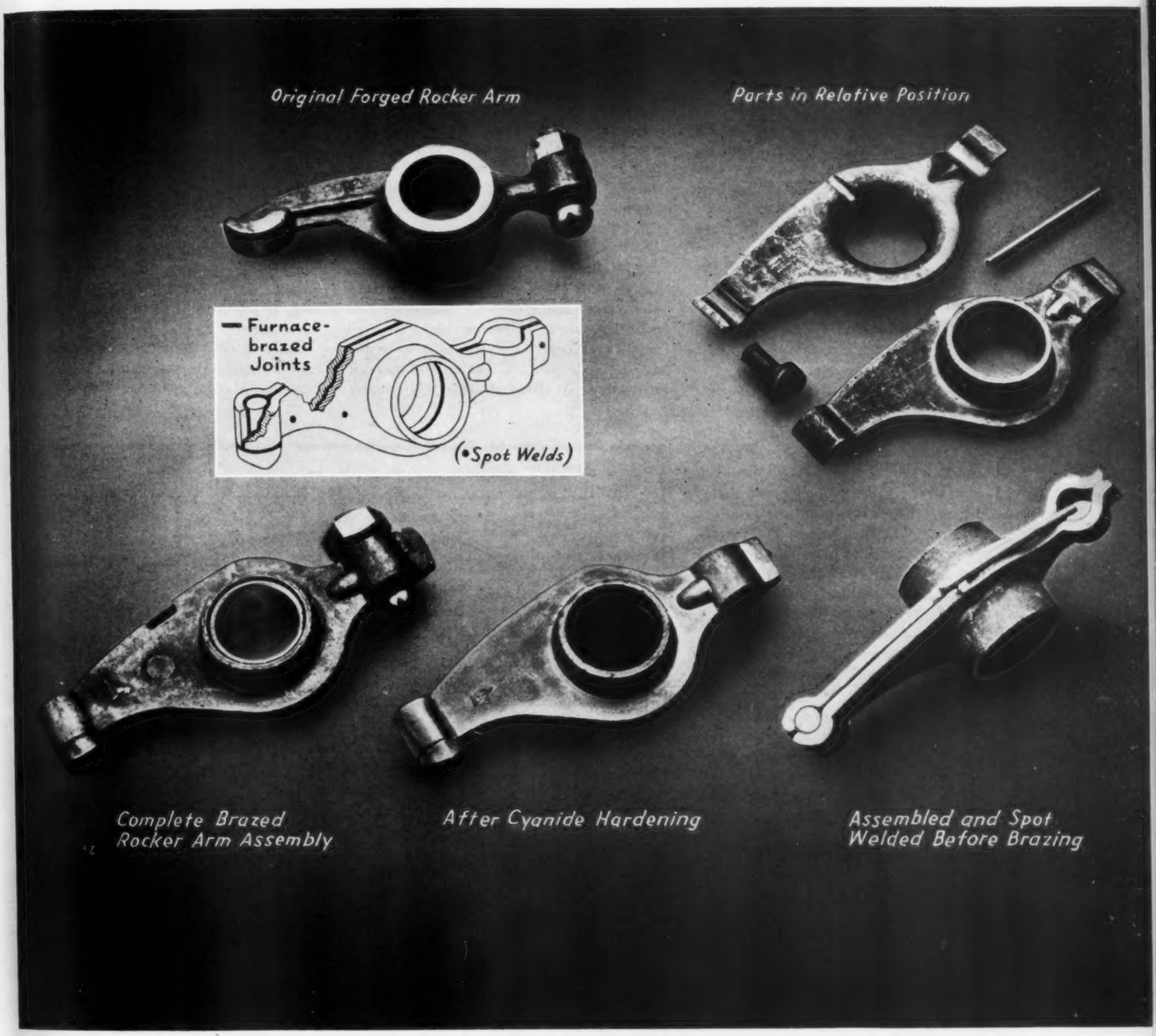
Brake toggle for tractors, originally cast, when redesigned for fabrication by furnace brazing consists of five steel stampings, a standard nut, and four pieces of copper wire. The component parts are shown above in their relative position. The three major parts are designed with stamped projections which locate them accurately during assembly. After the parts have been assembled they are spot-welded as shown in the diagram to hold the assembly together. Copper brazing wires are simply laid into position, two of them being bent into circles to fit around the built-up bosses. The entire assembly is then passed through a controlled-atmosphere electric furnace, where the copper melts and flows uniformly into all joints.

In this particular brake toggle part, punched holes are accurate enough for the service intended, so that no drilling is needed. Similarly, no grinding or finishing of stamping surfaces or brazing fillets is necessary. The standard nut eliminates drilling, reaming, and tapping operations formerly necessary on the cast brake toggle.

Weight of the brazed toggle is 9½ oz., whereas the weight of the cast part was 13½ oz. Savings in machining operations and in materials is claimed to have reduced costs by 46 per cent.

FURNACE BRAZING—I

L. SWANSEN, *Allis-Chalmers Manufacturing Company*



Furnace-brazed rocker arm assembly, redesigned from forged part, is made of six pieces; two steel stampings which form the arm, a screw-machine part which serves as the face bearing, a bushing and an adjusting screw with lock nut. Assembly operations consist of spot-welding the two SAE 1010 stampings together, inserting the high-carbon steel screw machine part, placing the brazing wire in position as shown, and passing the assembly through a controlled-atmosphere electric furnace operating at 2,060 deg. F. The entire assembly is then cyanide-hardened to add stiffness and to harden the face bearing. Final machining operations include tapping the hole for the adjusting screw and inserting and burnishing the bushing.

Principle aims of this fabricated design were to reduce machining formerly necessary and to overcome the chief difficulty of the forging in that the carburized and hardened ends of the forging were not uniform, and often failed because too soft or brittle. Result was the elimination of cleaning, boring, facing, milling, and drilling operations, and more accurate and uniformly hardened contact faces. Lighter weight of fabricated rocker arm permits use of lighter valve spring and push-rod.

HYDRAULIC FLUIDS—II

Fundamentals Affecting Circuit Efficiency

NORMAN E. MILLER
Consulting Hydraulic Engineer

THE IMPORTANCE of selecting a hydraulic fluid of correct viscosity, wear resistance and demulsibility was discussed in Part I of this article which appeared in the September issue of PRODUCT ENGINEERING. This part of the article will cover the significance of viscosity index, pour point, neutralization number, stability and miscellaneous properties.

Viscosity Index

In considering viscosity index, regarded as one of the more important specifications, it would be well to review the reason and meaning for this specification. Viscosity index is generally defined, as Dean and Davis Viscosity index, Dean and Davis being the names of the research investigators credited with developing this system of evaluating temperature viscosity change.

The actual purpose of viscosity index as such is to evaluate relatively the magnitude of viscosity change occurring through any given temperature rise. Oils of different bases, such as paraffin, naphthenic, asphaltic, have varying viscosity indexes.

At the time Dean and Davis were developing the viscosity index system, a certain paraffin base oil was found to change the least in viscosity when subjected to a given temperature rise. Hence, this was given a rating of 100 per cent, or said to have a Dean and Davis viscosity index of 100. All other oils investigated at that time had indexes of less than 100. Of course, the perfect oil would have no change in viscosity over a wide temperature range. Unfortunately, no such oil exists. All oils thin out, have lower viscosity, at higher temperatures; and thicken, increasing viscosity, at lower temperatures. However, as pointed out, these changes in viscosity with temperature are not identical, nor even nearly identical, for different bases of oil. Some may thin out rapidly and thicken quickly, others relatively slower. Hence, viscosity index, or V.I. as it is commonly termed, is a method devised to show the relationship between these viscosity changes of various oils with given changes in temperature. Therefore a high V.I. rating means the oil changes less in viscosity with temperature change than would an oil of lower V.I.

Since most hydraulic equipment depends for high pressures upon maintaining a viscous seal between moving parts, it is extremely important that wide fluctuations in viscosity be avoided. This is particularly so when one considers that on a typical hydraulic pump the overall efficiency can drop to a very low value, say around 25 per cent, and still the input power to the unit will remain as high as if the unit were developing 75 per cent or higher overall efficiency. To illustrate this relationship further, a typical example is recorded in Fig. 3. Here it is seen that the only reason for decrease in overall efficiency is lowered viscosity, the dynamometer used in the test indicated no reduction in power input. By disregarding the critical effect of viscosity index, this user was paying for efficiencies of 75 per cent or over, and was obtaining about one-third that amount because of the use of oil of the wrong operating viscosity.

Of course, there are other methods for controlling this loss in efficiency, such as the inclusion of heat exchang-

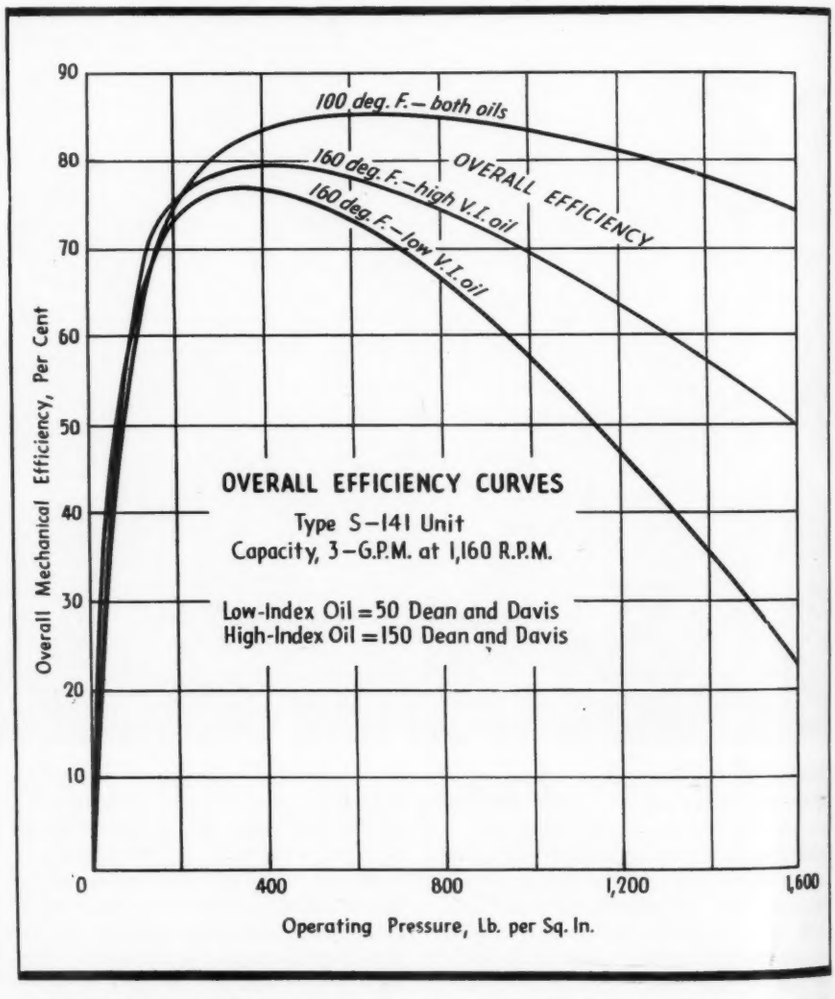


Fig. 3—Curves which show losses in efficiency of hydraulic units with high and with low viscosity index oils

Table I—Values for Calculating Viscosity Index by Dean and Davis Method

Viscosity — S. S. U.			Vis. Diff. L −H	Viscosity — S. S. U.			Vis. Diff. L −H	Viscosity — S. S. U.			Vis. Diff. L −H
At 210 Deg. F.	At 100 Deg. F.			At 210 Deg. F.	At 100 Deg. F.			At 210 Deg. F.	At 100 Deg. F.		
	Series H	Series L			Series H	Series L			Series H	Series L	
40	107	138	31								
41	120	161	41	81	810	1674	864	121	1643	3902	2259
42	133	185	52	82	829	1721	892	122	1665	3966	2301
43	147	210	63	83	849	1769	920	123	1688	4031	2343
44	161	237	76	84	868	1817	949	124	1710	4097	2387
45	176	265	89	85	888	1865	977	125	1733	4163	2430
46	191	293	102	86	907	1914	1007	126	1756	4229	2473
47	206	322	116	87	927	1964	1037	127	1779	4296	2517
48	222	353	131	88	947	2014	1067	128	1802	4363	2561
49	239	386	147	89	966	2064	1098	129	1825	4430	2605
50	257	422	165	90	986	2115	1129	130	1848	4498	2650
51	272	456	184	91	1006	2166	1160	131	1871	4567	2696
52	288	491	203	92	1026	2217	1191	132	1894	4636	2742
53	305	525	220	93	1046	2270	1224	133	1918	4705	2787
54	322	561	239	94	1066	2322	1256	134	1941	4775	2834
55	339	596	257	95	1087	2375	1288	135	1965	4845	2880
56	356	632	276	96	1107	2428	1321	136	1988	4915	2927
57	374	669	295	97	1128	2481	1353	137	2012	4986	2974
58	391	706	315	98	1148	2536	1388	138	2036	5058	3022
59	408	743	335	99	1168	2591	1423	139	2060	5130	3070
60	426	781	355	100	1189	2646	1457	140	2084	5202	3118
61	443	819	376	101	1210	2701	1491	141	2108	5275	3167
62	461	857	396	102	1231	2757	1526	142	2132	5348	3216
63	478	897	419	103	1252	2814	1562	143	2156	5422	3266
64	496	936	440	104	1273	2870	1597	144	2180	5496	3316
65	514	976	462	105	1294	2928	1634	145	2205	5570	3365
66	532	1016	484	106	1315	2985	1670	146	2229	5645	3416
67	550	1057	507	107	1337	3043	1706	147	2254	5721	3467
68	568	1098	530	108	1358	3102	1744	148	2278	5796	3518
69	586	1140	554	109	1379	3161	1782	149	2303	5873	3570
70	604	1182	578	110	1401	3220	1819	150	2328	5949	3621
71	623	1225	602	111	1422	3280	1858	151	2353	6026	3673
72	641	1268	627	112	1444	3340	1896	152	2378	6104	3726
73	660	1311	651	113	1466	3400	1934	153	2403	6182	3779
74	678	1355	677	114	1488	3462	1974	154	2428	6260	3832
75	697	1399	702	115	1510	3524	2014	155	2453	6339	3886
76	716	1444	728	116	1532	3585	2053	156	2478	6418	3940
77	734	1489	755	117	1554	3648	2094	157	2503	6498	3995
78	753	1534	781	118	1576	3711	2135	158	2529	6578	4049
79	772	1580	808	119	1598	3774	2176	159	2554	6659	4105
80	791	1627	836	120	1620	3838	2218	160	2580	6740	4160

The Dean and Davis method of evaluating the viscosity-index of an oil is a comparison of the oil under test with two standard oils, arbitrarily chosen so that all three oils have the same Saybolt Viscosity at 210 deg. F. The standard oils are given in the above table.

Series H oils are given a rating of 100 V.I. as they probably had the least viscosity change with temperature of any straight run oils known at that time.

Conversely, Series L oils are rated zero Viscosity Index. However, present day oils have a V.I. greater than 100 or less than zero. The viscosity index of an oil is determined by the following formula:

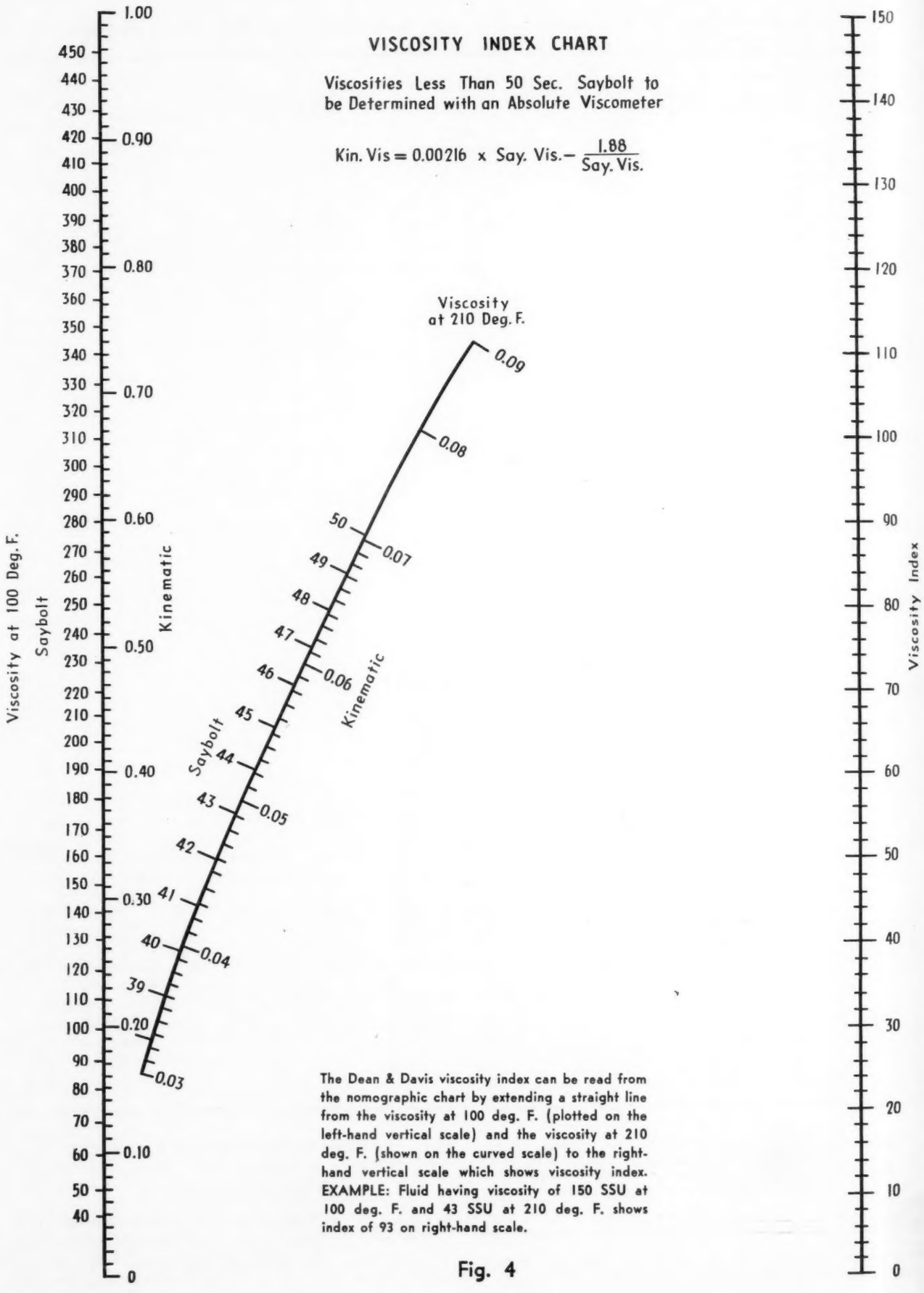
$$V.I. = \frac{L - U}{L - H} \times 100$$

in which L, H, and U are the Saybolt Viscosities at 100 deg. F. of a Series L oil, a Series H oil and the oil under test, respectively, all three oils having the same viscosity at 210 deg. F. This viscosity in above table determines the values of L and H to be used in the formula. *(Continued on next page)*

VISCOSITY INDEX CHART

Viscosities Less Than 50 Sec. Saybolt to be Determined with an Absolute Viscometer

$$\text{Kin. Vis} = 0.00216 \times \text{Say. Vis.} - \frac{1.88}{\text{Say. Vis.}}$$



The Dean & Davis viscosity index can be read from the nomographic chart by extending a straight line from the viscosity at 100 deg. F. (plotted on the left-hand vertical scale) and the viscosity at 210 deg. F. (shown on the curved scale) to the right-hand vertical scale which shows viscosity index. EXAMPLE: Fluid having viscosity of 150 SSU at 100 deg. F. and 43 SSU at 210 deg. F. shows index of 93 on right-hand scale.

Fig. 4

ers and thermal regulators in the circuit to hold viscosity at a predetermined value. However, this auxiliary equipment is an additional expense and a considerable part of the benefit derived therefrom may be obtained by using fluids possessing a high viscosity index. A further point in behalf of high viscosity indexes is the fact that slow down, following a cold start, is minimized. Often production in the entire plant will depend upon the continued operation of a certain production line, which may be driven with a hydraulic power unit, or a certain battery of production machines may be driven and controlled hydraulically. Consequently, where man hours are lost and unit costs pyramided with every minute of delay, it is vitally important that the hydraulic drive or control does not slow down with temperature rise, as it will unless preventive measures are applied.

For the convenience of the reader a viscosity index chart is shown in Fig. 4. This chart covers a wide range of fluids and indexes and may be used to check readily the viscosity index of ordinary hydraulic fluids. The only requirements are the viscosity values at the two temperatures, 100 and 210 deg. F. These are the standard temperatures used by most refiners in stating simple viscosity specifications. Where the viscosity index is desired with absolute accuracy, it is recommended that the data shown in Table I be used to compute the viscosity index. As explained in the notes accompanying Table I, the method of computation is relatively fast and simple.

In briefly summarizing this matter of viscosity index, it should be pointed out that an oil having a viscosity index of less than 60 will, generally speaking, contribute to loss of power and reduced overall efficiencies. There are many good fluids that can be obtained, that possess indexes ranging from 60 to 125 Dean and Davis. Broadly speaking, it is not always safe to say that the one that checks high will be better than the one checking low. However, the high one will be superior, speaking strictly from a viscosity temperature and hydraulic slip standpoint. If one attempts to make an overall comparison, considering index only, he is likely to go astray on such specifications as demulsibility, wear resistance, pour point, neutralization number, and similar specifications.

Pour Point

While hydraulically equipped machine tools are not usually thought of as being operated at sub-zero temperature, the pour point specification must nevertheless be taken into consideration.

There are several reasons why the pour point value must be considered carefully. The induction capacity of any hydraulic pump irrespective of make or excellence of design and workmanship cannot exceed a certain value. This value is, simply speaking, atmospheric pressure. In other words, no matter how excellent the pumping equipment is, in terms of induction ability it cannot do better than to develop a perfect vacuum on its intake side. Experience with hundreds of manometer tests has shown that 28 in. of mercury is about the maximum to be expected from the best hydraulic pumping equipment available. If the barometric pressure at sea level, 29.92 in. of mercury, is divided into atmospheric pressure at sea level, 14.75 lb. per sq. in., it is found that a barometric inch of mercury is equal to approximately 0.49 lb. per sq. in. pressure. Hence, a pump developing 28 in. of mercury vacuum on the intake side will have a maximum inductive capacity of approximately 13.7 lb. per sq. in. at sea level. Contrasted to this, the capacity on the pressure side of the pump is frequently around 1,000 lb. per sq. in. or roughly 73 times the amount available on the suction side.

The pressure drop through the induction system varies, in addition to other factors, approximately as the 0.5 power or square root of the Saybolt viscosity. When considering viscosity alone and its effect in terms of pressure drop through the intake system, it must be remembered that the viscosity increases rapidly at a much higher exponential value than the pressure drop exponent of 0.5. Therefore it is clearly important that the viscosity be kept as low as possible to prevent further inroads on the already limited induction force.

When low temperatures are expected in operation, it is of utmost importance that the pour point of the fluid be specified considerably below the lowest expected operating temperature of the pump and reservoir. The pour point should be much lower than the minimum expected operating temperature for the obvious reason that the fluid tends to thicken rapidly as the pour point is approached. The author has been successful in field experiences in maintaining a spread of about 30 deg. between the pour point and the lowest expected operating temperature. In other words, if an ambient temperature of 50 deg. F. is anticipated, the pour point of the oil should not exceed 20 deg. F. Where possible it is desirable to have an even wider spread, since the above condition was intended to state the minimum limit for temperature spread, and is contingent upon the use of fluids of correct operating viscosities.

It is possible with ordinary paraffin base mineral oils to lower the pour point considerably by the addition of certain hydrocarbon compounds such as Paraflow. The action of these compounds is to change the molecular structure of the wax content from crystalline to amorphous, in other words, to effect a lower film shear value between the various wax molecules. This is the same treatment given to the greater proportion of our crankcase lubricating oils known as SAE 10W, 20W, and so forth. Some oils before the addition of these compounds would be termed simply SAE 10 or SAE 20 lubricating oil.

Naphthenic base oils ordinarily have an inherently low pour point. The author has investigated oils of this base, with natural pour point values well below zero degrees Fahrenheit.

When extreme low temperatures are anticipated it has been found practical for some installations to blend the hydraulic oil with 10 or 20 per cent of sulphur free kerosene. Mixtures of SAE 10W oil and 20 per cent of well refined kerosene have been used successfully in certain outdoor applications such as hydraulically equipped snow plows and other road machinery.

Summarizing the effects of pour point, it has been found that it is necessary to maintain a spread of 30 deg. between the pour point and the lowest expected operating temperature. Failure to observe pour point specifications may cause serious cavitation of the pumping equipment resulting in excessive noise, and aeration of the oil reservoir contents with all the attendant evils accruing therefrom.

Neutralization Number

The term neutralization number is intended to define the relative degree of acidity present in any mineral oil, or in certain compounded hydraulic fluids. The neutralization number is expressed as a decimal, for example, 0.08 and 0.10, the decimal indicating the fractional part of a milligram of caustic, usually potassium hydroxide, KOH, required to neutralize one gram of the oil. Such a test indicates in mineral or compounded hydraulic fluids the presence of organic constituents having acid characteristics and the contamination by alkalies or mineral acids. The presence of acid in the reservoir contents of any hydraulic system will rapidly increase the tendency to sludge or to emulsify. In addition to the accelerated emulsification that occurs, corrosion and pitting of bearing surfaces often develop, resulting in excessive wear.

Fatty acids, sometimes termed saponifiables, are present in animal or vege-

table oils and therefore in oils compounded with these products. These are definitely harmful if present in anything except minute quantities. They react with metals to form metallic soaps and other compounds which accelerate sludging and emulsification.

Mineral acids such as sulphuric acid, or alkalies, are infrequently found in well refined hydraulic fluids. When present, such constituents usually indicate that refining or solvenizing chemicals have not been removed entirely. This is one reason for making sure that, when new, hydraulic fluids have a neutralization number of 0.10 or less. Where an unusually complex or expensive circuit is involved, it is advisable to have the maintenance department check neutralization numbers periodically. Any increase in neutralization number during service should be very gradual. An abrupt increase is generally a pretty fair barometer of forthcoming trouble.

Stability

The stability of hydraulic fluids in resisting oxidation and deterioration in a hydraulic system operating at above normal temperatures, frequently in the presence of condensation or water, is an important factor to observe in the application of the proper fluid. Even the finest petroleum products, refined by the best methods known to the industry, will oxidize and deteriorate rapidly under certain conditions. When subjected to elevated temperatures in the presence of certain metals, water, and moisture, all mineral oils undergo oxidation, the various hydrocarbons of which they consist breaking down to form non-lubricating products and corrosive acids. This deterioration is greatly accelerated when accompanied by high operating temperatures and/or pressures. However, certain oils tend to oxidize more than others.

There are in use throughout the oil industry a number of methods to determine the resistance of an oil to oxidation. However, correlation of the results obtained in the laboratory and actual service results have never proved an unqualified success. It can, however, be stated that generally speaking, although not always true, oils having natural viscosity indexes obtained without index additives are more stable than those having lower indexes, or high indexes if obtained by the only presently known index additives. A further point worthy of mention in connection with stability is that the presence of air in the circuit brought about by cavitation, high velocities, jet effects, and other causes, will accelerate and aggravate the tendency to oxidize. Excessive oxidation is usually accompanied by increased acidity and sometimes a precipitation of resinous or asphaltic materials known as sludge. This is particularly so in instances where abnormally high operating temperatures prevail.

Miscellaneous Properties

Among the miscellaneous properties of a hydraulic fluid, probably the most important is compressibility, particularly so when considered from a circuit performance standpoint. Little is heard of this compressibility angle, perhaps because there is so little known about it, certainly not because it is not of appreciable significance. There was for years a widely accepted rule of thumb definition of oil compressibility, stating that a column of oil 5 ft. in length would compress one-sixteenth of an inch under a pressure of 1,000 lb. per sq. in. Actual measurements have shown this assumption to be woefully wrong. The compressibility in the first place, varies considerably with temperature and base origin of the oil. The so-called premium or Pennsylvania paraffin base oils having from 10 to 20 per cent greater compressibility than the Mid-Continent and Coastal bases.

In considering the practical effects of oil compressibility it must first be recalled that tool chattering, hydraulic hammer, poor control, are each closely related to this property of elasticity or compressibility. When one considers that an average well-refined widely sold hydraulic oil has a modulus of elasticity of 300,000 at normal operating temperatures, the relationship assumes its proper importance. In other words, the average oil elasticity as scientifically calibrated is now known to be approximately 100 times greater than that of steel, which is commonly thought of as having a modulus of elasticity of 30,000,000. One must here bear in mind that the elasticity varies inversely as the modulus.

Significant though, fluid compressibility has been with us all through the years and limited data pertaining to it have been published in the International Critical Tables.

Field experiences consisting of perplexing and hard to trace operating characteristics have resulted in recent investigations concerning compressibility and the successful development of remedial measures for overcoming the effects.

In covering other miscellaneous properties, it should perhaps be pointed out that the Flash and Fire Specifications were purposely ignored because of the fact that flash and fire specifications are important chiefly in determining the degree of refinement and care used in the production of crankcase oils. The operating temperatures in a hydraulic system should be controlled to levels sufficiently low as to minimize the effect of fire and flash as having an important bearing upon the overall operation. In addition to this, many hydraulic systems are operating on what is commonly termed a "closed circuit" wherein the oil is not subjected to the open atmosphere and therefore volatility, of which the flash and fire tests are a measure, is unimportant.

Conclusions

It is hoped that the equipment user and equipment engineer will realize that although there are many pitfalls for the uninitiated, the selection and proper application of hydraulic fluids is a practical and readily accomplished procedure.

FIRST, the manufacturer's recommendations should be adhered to carefully by the user.

SECOND, in selecting the source for obtaining the correct fluid the reputation and integrity of the supplier should be scrutinized closely.

THIRD, there are available at the present time a considerable number of excellent hydraulic oils and compounded fluids that fulfill every condition covered herein.

It is also unfortunately a fact that the subject of hydraulic fluids has not as yet been broadly nor sufficiently well understood to attain justifiable attention. Yet throughout industry, there are few places where such appreciable increases in output and reductions in power consumption can be attained so readily and economically, as in the application of correct fluid medium to hydraulic power transmission systems.

Current Quotations

"It is a fallacy to believe that we have reached the end of the road of progress. The frontiers of science and industry ahead hold promise of new jobs and new improvements in the standard of living surpassing even the progress of the past which has made this the most envied nation on earth."—DR. CHARLES F. KETTERING, *V. P. in charge of Research, General Motors.*

News and Summaries

Rock Island Arsenal Tank Designs Letter-Perfect, Say Chrysler Officials

Rumors that mistakes necessitated redesign and delay of several weeks are false, say the engineers who are building the tanks

RECENT RUMORS have it that when Chrysler Corporation engineers looked over plans for government-designed 25-ton tanks, they found so many mistakes they had to spend weeks correcting them before they could get set for production on defense orders. The truth of the matter, according to what Chrysler officials recently told PRODUCT ENGINEERING, is that the government blueprints were absolutely perfect. Chrysler officials had nothing but the highest praise for the design work done at the Rock Island Arsenal, and stated that the tank was surprisingly smooth-riding.

Birth of the rumor probably was in the fact that 197 Chrysler engineers worked for four weeks, plus overtime, to make a complete production study of the tank. This work involved no redesign but confined itself to determination of types of machine tools and equipment required, and plant layout.

Blueprints of the tank received by Chrysler totaled 186 lb., and included more than 6,000 detail drawings and sub-assemblies. These blueprints, the only complete set, were immediately sent out to the pattern shop where each part was made of wood exactly to specifications, painted and lacquered. The finished parts were then sent to an assembly room where they were put together. So accurate were the Rock Island blueprints that the complete assembly was made without even using a file on any of the parts.

Design methods used at Rock Island Arsenal are such as to immediately catch and correct any errors in drawings. When the first set of blüeprints is completed, they are sent to the shop where parts are made under rigid inspection, exactly to blueprint specifications. These parts are then assembled and if any difficulty is encountered the blueprints are immediately checked.

Necessary changes and corrections are made directly on the drawings, and corrected blueprints are then sent to the shop to have new parts made. No changes in parts are made except according to blueprint specifications. If a part doesn't fit at first, the assembly man cannot file it down to a size which does fit, unless there is a blueprint specifying the changed dimensions. Thus, for every correct part or assembly, there must be a letter-perfect blueprint.

All-plastic airplanes, although prematurely publicized, are bound to "arrive" some day, the experts predict. Timm Aircraft Corporation recently got off to an early start when they announced their "Aeromold" all-plastic training plane, pictured above, arousing considerable interest in the aircraft industry. Army and Navy officials have been looking it over. The only reason why plastics have not been used more extensively in fuselage and wing construction, according to authorities, is that engineers have not yet compiled enough stress data. The strength of laminated plastic airplane sections is not yet fully known, and will not be known until considerable more research has been done. Consequently the plastic plane designer, starting from scratch, must add a little more strength than is necessary in order to be on the safe side.

Steam-Powered Airplane Still a Possibility

STEAM POWER FOR AIRCRAFT is still an interesting possibility, many reliable authorities state. The first airplane ever to leave the ground was powered by a small two-cylinder steam engine built by Clement Ader in 1890 in France. Recent reports from Europe indicate that Germany is now hard at work perfecting a steam power plant for airplanes, and in this country a small airplane has been flying around under steam power for some years.

Several years ago Besler Brothers built a light-weight steam engine using a continuous cold-type steam generator, and flew it successfully in a small biplane. This trial was merely experimental, and the condenser capacity was reported inadequate for full-throttle operation. Besler Brothers own many of the old steam automobile patents and have built several steam cars.

In 1933, the now extinct Great Lakes Aircraft Corporation designed a steam airplane plant, using a special LaMont-type high pressure boiler. The boiler was to be located in the fuselage of the plane, with two 18 x 36-in. turbines, turning at 22,000 r.p.m. and geared down to propellers rolling at 700 r.p.m., installed in the wings. Each turbine would turn out about 1,500 hp. LaMont claims to have solved the condenser.

The Great Lakes airplane was never

built, but many of the paper-facts are extremely interesting. The fuel efficiency, based on weight, would have been about the same with this steam engine as with a comparable gasoline engine, in comparing high-heat fuel oil with high-octane gasoline. The plane, if ever built, possibly would have surpassed the clipper ship of its day on pay load, speed, fuel economy, climb, and altitude.

Steam flying may be only a pipe dream. At any rate, the gasoline engine has put the steam engine pretty much on the shelf. Gasoline engines are at their best on constant throttle, as in airplanes. Where the steam engines would really shine, the experts say, would be in automobiles, which require variable power. Development of steam power for airplanes in this country is now practically at a standstill. However, there is pressure continually being put on the government in an attempt to get some money from the Military for development. Aim of those who claim to be in the know is for a steam power plant which would be suitable for bombers and other large aircraft where power requirements are considerably in excess of 2,000 hp. Military experts are a bit doubtful about the advantages of steam power for airplanes. They are afraid that the condenser might freeze, or that bullet holes will cause considerable trouble with this type of a system.

Sound Film Describes Design of Steel Castings

PRINCIPLES of steel casting design are brought out in an interesting and educational manner in a new sound slide film available from the Steel Founders' Society of America. The film runs for half an hour, and is described by the Society as containing probably more information on this important subject than has been accumulated anywhere else. Various principles of design are simply and clearly brought out, using data contributed by members of the Society.

Thin Film of Barium Serves as Lubricant

MOST EFFICIENT LUBRICANT for a special type of X-ray tube, in which the anode target is rotated at high speed, is a thin coating of metallic barium, report three General Electric engineers. In the high vacuum demanded by X-ray tubes, ordinary lubricants cannot be used since vaporization of the lubricant spoils the vacuum.

The anode target in this X-ray tube is rotated at about 3,560 r.p.m. by induction. In previous tubes, the bearings of this target were unlubricated. In research conducted by Zed J. Atless, Jack T. Wilson, and James C. Filmer, of General Electric X-Ray Corporation, it has been found that a number of metallic films have possibilities as lubricants in this application, but barium has proved to be the most successful. The metal is applied in a thin film on the steel ball bearings of the target by vaporization. After applying the barium film, the coasting time of the target was increased from 12 sec. to 8 min. Under normal operating temperatures of the X-ray load, the film will effectively lubricate for up to 100 hours.

New Standards Published Covering Twist Drills

TWIST DRILL SIZES, long a controversy between manufacturing and user groups, have finally been standardized, the American Standards Association reports. The new standards recommend the adoption of 116 drill diameters ranging from 0.0156 to 0.5000 in., inclusive, as standard.

These drills are used throughout the mechanical industries from automobile manufacturers down to the smallest machine shop, and by the casual home mechanic. The recommended standard has been planned to fill all of the normal needs of these users.

Do You Know That—

V-BELT DRIVES using larger diameter sheaves develop higher speeds, require fewer grooves and fewer belts—therefore may actually cost less than drives with smaller sheaves. (27)

ARMOR PLATE for war tanks can be welded, but welding softens the heat-treated plate at the seams, leaving a veritable "Achilles heel." Hitler didn't find this out until he invaded Poland when the Polish army began to shoot holes through his welded tanks. (28)

MACHINE PARTS may become slightly magnetized when the machine is running because of mechanical impact and vibration which orient the molecules of the part in the earth's magnetic field. Often these magnetized parts serve the useful purpose of removing minute but harmful iron particles from the lubricating oil. (29)

Turbine steel rods, imprisoned in these furnaces for more than eight years as a part of General Electric's investigations on creep strength, are here being checked over by engineers E. L. Robinson and E. A. Smith. Creep strength of a given steel has recently been found to depend largely on grain structure through heat treatment, reports H. L. Weaver. For high creep temperatures the optimum grain size (size of ferrite grains, patches of pearlite or Widmanstätten areas) is large, and for low temperatures the optimum grain size is small. Uniformity of grain is also important. This discovery explains why two steels of same analysis do not always have same creep strength, and will enable the designer to specify turbine steels with more confidence.

ENGINEERING IN WASHINGTON

McGRAW-HILL WASHINGTON BUREAU, *PAUL WOOTON, CHIEF*

Underground airplane plants and military and civil hangars are much in discussion. Washington engineers are doubtful about economic feasibility of earth bomb-proofing. One highly placed authority said that the underground space for a plane would cost many times as much as the plane itself, so you could lose a few ships by bombing in the open, and still be ahead. It is estimated by some that about 30 feet of earth would be necessary to protect against sizeable bombs. The weight of such a covering would require immense and expensive support. The Germans have some subterranean airplane works, but no one knows how many.

2,000 or 3,000 airports are wanted by the Administration and the Army and Navy, in addition to our present system of fields. Congress is considering an appropriation of half a billion dollars for the purpose. Aviation is just learning that the biggest, best-planned air terminals can handle surprisingly few traffic movements. Therefore, if it became necessary to maneuver a force of 50,000 planes in this country there would be congestion. If we are to build thousands of ports, engineers say, we must find some way to use treated earth surfaces, except in major termini, because hard runways are too expensive. Furthermore, port engineers say, the airplane designers will have to cooperate. So far they have designed for speed and fuel economy, giving little attention to quick takeoff.

Graduate engineers who would like a defense job but whose training isn't in the specialized lines now in demand should keep in touch with plans for a big program of short courses to be offered in engineering schools around the country. Idea is that a mechanical engineer, for instance, can take an intensive four-months course in aircraft engines and qualify for a job with a plane manufacturer. President has asked Congress for $9,000,000 to pay for the cost. No tuition will be charged; enrollment will be handled through the schools themselves. Each engineering school will offer courses in the specialties needed in its neighborhood. Typical specialties in which shortages exist or are expected include: naval architects, ship draftsmen, and marine engineers; airplane structural, engine, and instrument experts; material inspectors and tool and machine designers.

Invisible airplanes are being used by the British, the Nazis charge, so as to make anti-aircraft gunning difficult. The British reply that this charge is a poor excuse for bad marksmanship. Bureau of Standards experts confirm what you probably suspect: that there is no such thing as paint that will render a surface invisible. What the British did was probably a very skillful job of camouflage. It is possible to use color and line to conceal almost completely an object in a given background. Trouble is, you can't depend on the background staying the same, especially the sky.

Alcohol motor fuel is an old story but it might be news some day. The Department of Agriculture's new laboratory at Peoria, Ill., is going to work on industrial uses of corn, using the full time of 12 technicians. A complete alcohol pilot plant is being installed. Last year the United States consumed 22 billion gallons of gasoline. If ten per cent of that had been corn alcohol, 900,000,000 bushels of corn would have been consumed. The trick, you see, is to pass some of the rich oil industry over to the farmer, who is not so rich. Fuel alky can be made of wheat, white and sweet potatoes, and other vegetation, too. It's a big subject. You can learn more by writing to the Superintendent of Documents, Washington, D. C., for "Motor Fuel From Farm Products."

Radium control instruments, which will be valuable to radium and high-voltage ex-ray technicians, have been developed by Dr. Leon F. Curtiss, and others, in the Radioactivity Section of the Bureau of Standards. The trick is done through a circuit for integrating pulses from tube counters, designed by Dr. Curtiss. One instrument measures low gamma ray intensities down to the equivalent of 0.1 microgram of radium. The other is a dosage meter for gamma rays, measuring in terms of roentgens per day, and giving an audible alarm.

Suicide boats, without suicide, are under experiment by the Navy. Chris-Craft shipped four "unsinkable" speedboats to Navy, to be equipped with controls that can be actuated by radio from airplanes. It is understood the boats will go later to Langley Field, Va., laboratory of the National Advisory Committee for Aeronautics. The apparent intention is to turn these lethal mavericks loose with torpedoes aboard, and guide them to the target from aloft. Radio-controlled bombing targets would be valuable to the Navy, too. The boats have a top speed of 35 m.p.h. and are 34 ft. long.

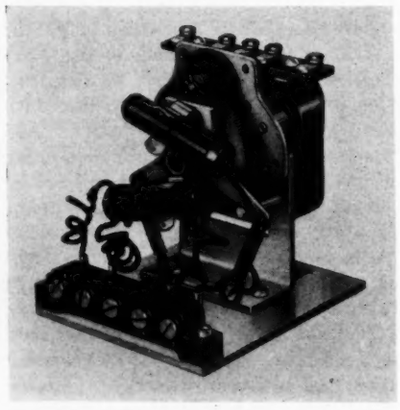
Merchant ship building might be expected to be curtailed as the Navy gets under way with its vast construction program. But officials of the Maritime Commission tell PRODUCT ENGINEERING that most commercial building is in yards that are not and will not be taking Navy work. Merchant bottoms that are now building in yards where Navy building is also in progress will be completed and no more will be laid down. They say that new ways can be added to commercial yards to take care of the Commission's 50-ships-a-year program as planned. There are several World War yards still unused, but engineers figure that changes in construction methods make it cheaper to build new yards rather than rehabilitate the old ones.

Talk about tin will confuse a lot of folks from now on. Don't feel inferior if you get mixed up yourself, because Washington is too. There are top-flight authorities who say our tin supply is safe enough, and we don't need smelters in this country. Others think smelters are imperative. Some politicans are more interested in grabbing smelters for their home towns than in the real problem. There are few, if any, persons in this country who know how to smelt tin, so we have sent for foreigners who do. The fantastic (but somewhat doubtable) tin king of Bolivia, Simon I. Patino, is here, dickering with the Defense Commission. The smelting industry may build tin plants, or the government may build them. Best guess is that at least one plant will be built.

New Materials and Parts

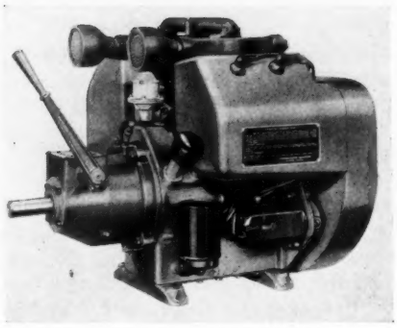
Motor Relay

Consisting of a small reversible, geared head, shaded pole Barcol motor which tilts two mercury tubes, this new Motorelay permits throttling, floating, or proportioning control of a 5-amp. load from a sensitive instrument having a current rating as low as 0.35 amp. The tubes are arranged to give single pole, double throw operation, with spring return to a neutral "off" position. Chattering is eliminated because a momentary contact at the instrument is not sufficient to close the mercury tube switch. Particularly suited to floating or proportioning control of motors operating large dampers, large valves, speed adjusters, voltage regulators and similar equipment. Barber-Colman Co., Rockford, Ill.



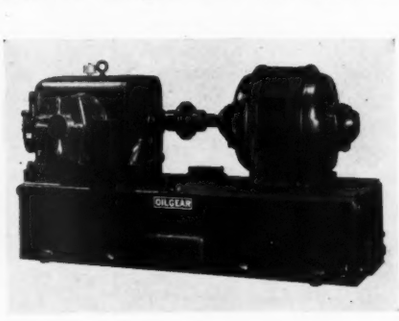
Heavy Duty Air-Cooled Engines

Type VE-4 air-cooled gasoline engine delivers a maximum torque of 600 in.-lb. at 1,600 to 1,800 r.p.m. and is rated at 20 hp. It is a V-type four-cylinder engine of compact design having a length of 24 in., height 23 in., width 21½ in. overall. The engine has a 3-in. bore, 3¼-in. stroke, and weighs 285 lb. The main bearings are Timken roller. Positive cooling is obtained from a large fan cast in the flywheel and efficient cooling is claimed even at extremely high temperatures. Cylinder blocks are cast separate from the crankcase. Removable valve seat inserts are used on both intake and exhaust. Major parts may be had of aluminum at extra cost with a reduction in weight of approximately 50 lb. Wisconsin Motor Corp., Milwaukee, Wis.



Variable Displacement Pumps

Two-way, three-position variable displacement pumps have internal pumping mechanisms for normal working pressure ratings of 1,100, 1,700 and 2,500 lb. per sq.in. and for peak pressures up to 3,000 lb. per sq.in. These new heavy-duty, radial, rolling piston pumps are arranged for flange mounting to a separate reservoir base, or to a reservoir integral with the machine.



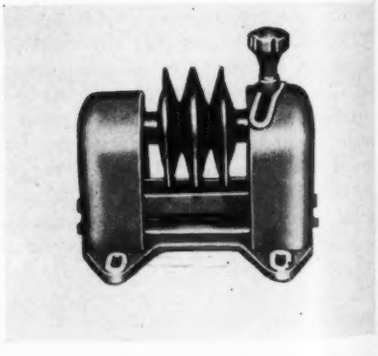
They are equipped with built-in auxiliary gear pumps which supercharge the high pressure unit for operating the pump control mechanism and for auxiliary purposes. Relief valves protect machine and pump against overload. Electric control makes it possible to reverse the direction of oil flow and also to select neutral position at will from a remote pushbutton or switch station. Type DX displacement pumps are manufactured in conventional sizes from 2 to 150 hp. Oilgear Co., 1301–1417 W. Bruce St., Milwaukee, Wis.

Gasoline Motor

Known as Model "ZZ", this new 4-cycle, air-cooled gasoline motor is rated at 6 hp. Piston displacement is 22.97 cu. in., with a 3 x 3¼ in., bore and stroke. Specially designed dust- and moisture-proof, high tension, flywheel magneto supplies ignition. Other features include Silchrome exhaust valve with alloy steel inserted seat, inserted molybdenum alloy valve guides, specially designed float feed type carburetor, adjustable mechanical governor, drop-forged counterweighted crankshaft, aluminum alloy pistons and connecting rods, pump and splash lubricating system, gasoline filter, oil-bath air cleaner, screened blower housing. Recommended speed range, as recorded in dynamometer tests, is 6.15 hp. at 2,200 r.p.m., 7.20 hp. at 2,700 r.p.m., and 7.70 hp., at 3,200 r.p.m. Also available in a light-weight model, with many parts made of cast aluminum instead of cast iron. Briggs & Stratton Corp., Milwaukee, Wis.

V-Belt Transmission

Known as the All-Speed, this new speed selector operates on the variable pulley principle, and consists of two interlocking pulleys mounted on a common shaft. The variable speed spindle can be pivoted laterally between the motor and the driven machine to provide new running diameters on the interlocking pulleys. Selected speed is



maintained by means of a lock nut on the control quadrant, and the oblique slot which automatically and positively guides the spindle at all times insures perfect belt alignment and increases belt life. This is a distinct advantage when the control unit is mounted vertically. Special impregnated bronze bearings are used, with forced lubrication provided at all times. Available in three sizes for ⅜ in., A, and B belts, and in speed ranges up to 12–1. Standard Transmission Equipment Co., 416 W. 8th St., Los Angeles, Calif.

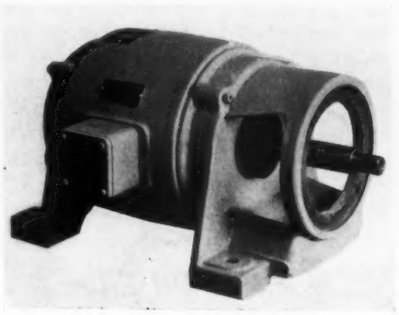
Roller Plunger Actuator On Die-Cast Micro Switch

Compact die-cast Micro Switch is now available with a roller type plunger

and a sturdy castellated brass bushing which mounts the roller either longitudinally or cross-wise. Accurate concentricity of the hardened and ground roller, which is carried on an oilless bearing, assures precision performance. Adjustment for actuation by cam or slide in either of the two castellated positions is quickly made by removal of the bearing screw. Micro Switch Corp., Freeport, Ill.

Pump Mount Motor

New line of motors with standard dimensions is designed for easy mounting of centrifugal and similar pumps. These motors are ball bearing with ratings from ½ hp. to 40 hp. for operation on 3-phase current at two, four and six pole speeds. Extended shafts and machined fits for the pump mounting are all standard and the same dimensions throughout the entire range of ratings. A large neoprene washer on the shaft between the pump packing and motor bearing, together with an ingenious trough cast around the bearing, make it impossible for any of the material pumped by the unit to find its way into the motor bearing. Projecting shaft is covered with a hard stainless



steel tube which is ground and polished. All units are carefully checked so that perfect alignment is assured. Sterling Electric Motors, Inc., Telegraph Rd. at Atlantic Blvd., Los Angeles, Calif.

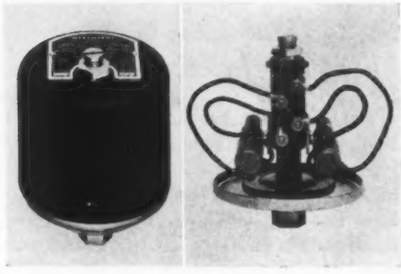
Drum Type Master Switch

This new drum type master switch, for 3-wire control on machine tools and other equipment, affords functions equivalent to two and three button heavy duty push button stations. By removing a stop post, the drum can be

converted from non-reversing to reversing. Available for surface, cavity, or panel mounting, and with standard or pistol handle grip. Ratings are for pilot circuits up to 600 volts a.c. and d.c. Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee, Wis.

Pressure Switch

Automatic pressure responsive switch opens or closes an electrical circuit under predetermined low or high pressure conditions. Designed for service on non-corrosive liquids or gases, these switches are fitted with Durakool mercury tubes which are inclosed in an unbreakable steel shell covered with Bakelite. False stops and starts caused by chattering are eliminated by a quick yet gentle tripping action imparted to the tube. Diaphragm is of pure gum rubber with duck insert for additional strength. Cut-in pressure ranges from 10 to 40 lb. per sq.in. cut-out pressure



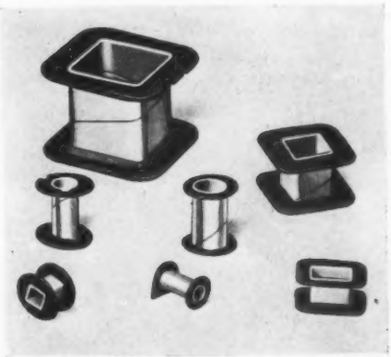
ranges from 20 to 70 lb. per sq.in. Pressure between cut-in and cut-out is variable between 10 lb. per sq.in. and 60 lb. per sq.in. Center & Giveen Co., 2024 University Ave., Berkeley, Calif.

Iron Powder

Iron Powder which is the equivalent of Swedish iron powder is now available as an American material. Because of the European situation, the supply of Swedish powder has been greatly reduced. In fact, none has come through for some months, so that the production of this important product here in America has become vital to this country's self-sufficiency. Metals Refining Co., Div. of Glidden Co., Hammond, Ind.

Coil Bobbin

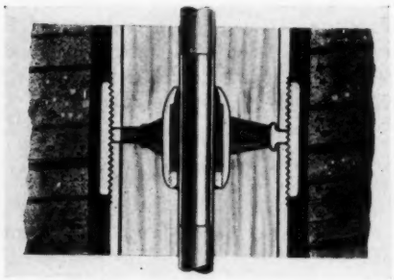
Permitting the use of smaller and more efficient bobbin coil rather than larger layer wound coil, these new bobbins are made of either Kraft or Fish paper, or a combination of both, depending on the requirements. The flanges are of vulcanized fiber, die-cut



to exact size and shape and pressed over the ends of the spirally wound tubes. Ends of tubes are swaged, locking the flanges in place. The bobbin is impregnated with a special lacquer which increases its strength and forms a seal between the tube and flange. Available in round, square or rectangular bobbins of any size with a high degree of accuracy. Precision Paper Tube Co., 2033 Charleston St., Chicago, Ill.

Water-Lubricated Bearing

Perfected for use in deep well vertical pumps, the pump shaft in a new type water-lubricated bearing is armored with chromium plating to a thickness of 0.003 in. over a Monel stainless sleeve every 10 ft. These sections of the shaft rotate within resili-



ent cutless rubber journals which are fluted. Any sand in the water flushes through the bearing. Because of the extreme hardness and thickness of the chrome coating, and its bearing contact with cutless rubber, no abrasion occurs. Peerless Pump Co., 301 W. Ave., 26, Los Angeles, Calif.

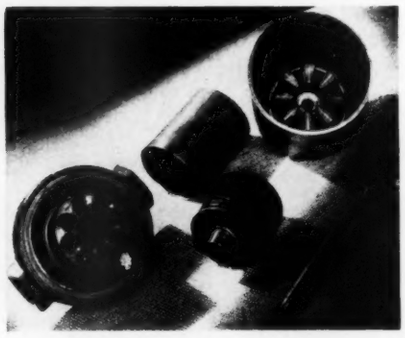
Leather "U" Packings

Designated Double "U" because they are doubly reinforced against wear on the contact leg and against pinch from excessive clearance, these new packings are available in section sizes from ½ in. to ⅞ in. for any plunger diameter from 9 in. to 90 in. Can be reinforced with any adaptable material over the "U" packing base, standard reinforcement being tough chrome tanned belting

leather backed by oak tanned belting leather. Patented phosphor bronze spring spacer maintains a light pressure against the inside of the bevelled contact lip near its pressure edge, insuring a primary seal until the packing lips can be set to the plunger surface under pressure. White oak spacer is steam bent to the packing channel, extends beyond the packing lip, and serves by its contact with the end of the packing recess to hold the packing in position. Wayne Davies Packings, Inc., 325 W. Huron St., Chicago, Ill.

Magnetic Oil Filter

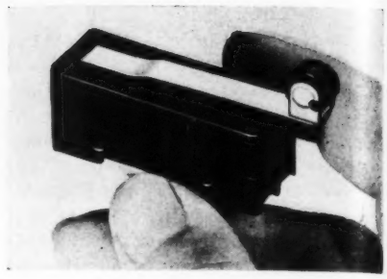
Powerful magnetic separator for use on lubricating and hydraulic oil systems, uses permanent magnets to extract minute iron particles that would pass through ordinary oil filters. The new filter, which permits an unobstructed flow of lubricant, consists essentially of a two-piece case in which is mounted two powerful Nipermag magnets. Between these magnets and within a bronze tube which directs the flow of



oil, are a series of grids made of a high permeability alloy. Magnetic particles of iron flowing into the filter are held either on the magnets themselves or on the large grid area. Cleaning is accomplished by disassembling and washing in kerosene, an operation which is greatly facilitated by the fact that the grids become non-magnetic when removed, thus permitting easy removal of iron particles. Known as the Ferro Filter, this new device has found extensive use during the run-in period of engines, and is also adaptable for cleaning lubricating oil in reduction gears and in certain ceramic applications. S. G. Frantz Co., 161 Grand St., New York, N. Y.

Spring Leaf and Roller Actuator on Mu-Switch

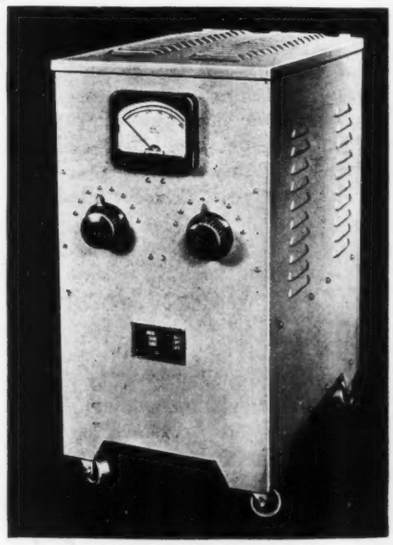
Comprising a strong phosphor bronze leaf spring securely riveted at one end to the Mu-Switch cover, and having at



its free end a ⅜ in. x ¼ in. roller made of graphite-impregnated, canvas-base bakelite, this new style actuator adapts the Mu-Switch to high-speed cam operations. The self-lubricated roller permits extremely close bearing tolerance with resulting accuracy of cam-controlled action, and gives quiet operation and freedom from wear or scoring of the operating cam. Type W actuator gives protection against over travel, and is claimed to have long life under conditions of high speed and high temperature. Mu-Switch Corp., Canton, Mass.

Portable Rectifier

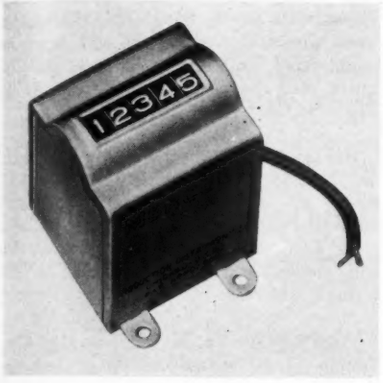
Designed especially for testing purposes, this new model SP-20 rectifier utilizes two mercury vapor tubes to obtain full wave rectification of single phase a.c. Output voltage can b adjusted by means of knobs on the front panel in steps of 32, 110, 115, 120, 210, 225, and 240 volts. Current capacity is 30 amp. at 32 volts d.c.; 20 amp. at other voltages. Output voltage remains practically constant from no load to



25 per cent overload. The rectifier contains no moving parts. Tubes have long life and operate at high efficiency. Supplied with or without a filter choke. Mellaphone Corp., Rochester, N. Y.

Electric Counter

Small, compact electric counter, known as the Mercury counter, requires only 2 watts operating energy, which insures long life of switch contacts and makes the instrument particularly suited for photo electric eye operation. Tamper-proof construction, non-reset feature, totally inclosed assembly, small size, convenient mounting, and attractive appearance make Mercury counters



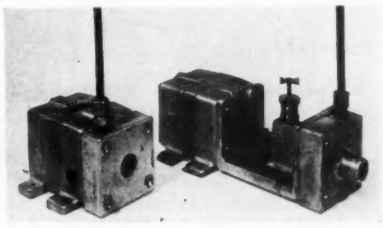
ideal for built-in applications. Case and frame are cadmium-finished steel with unbreakable plastic windows. Large white figures on black wheels are located close to the window for easy reading. Tested at 600 counts per min. Production Instrument Co., 702–10 W. Jackson Blvd., Chicago, Ill.

Tungsten Contacts

Electrical contact points made from fused pure tungsten metal feature uniformity in grain structure which eliminates any excessive oxidation or pitting during operation periods. The purity of this metal prevents film coatings from adhering to contact surfaces during the time when apparatus is not operating or during shut-down intervals. Electrical contact surface resistance is maintained at a minimum and long service operation is assured. Metroloy Co., Inc., 57 E. Alpine St., Newark, N. J.

Small Rotary Pumps

Small rotary pumps are a new development for light industrial applications and for laboratory use. For pumping different kinds of liquids, the pumps can be furnished in various alloys. They are efficient, run extremely quietly, and are coupled with a shaded pole type electric motor, which gives trouble-free continuous service. The sump style of pump, built integrally with motor, for use when located in sump well of oil, has dimensions of $4\frac{1}{2} \times 3\frac{1}{2} \times 4$ in. The gland style of pump, with flexible self-aligning coupling between motor and pump proper, has ground pump shafts,



large bearing surfaces, aluminum alloy housing, stainless steel impeller blade and shaft, and has dimensions of $8 \times 4 \times 3\frac{1}{2}$ in. Speedway Mfg. Co., 1834 S. 52nd Ave., Cicero, Ill.

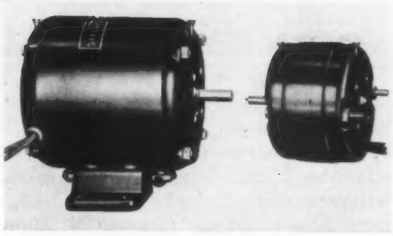
High Torque Motor

New high torque capacitor motor, with the necessary capacitors and centrifugal switch mounted within the

motor brackets, eliminates additional mounting and wiring costs. Waste packed sleeve bearings and dual voltage leads are standard and the entire unit is of the protected type of design. Known as Type CJ motor, it is available from ¾ to 3 hp. inclusive. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Shaded Pole Motors

New line of shaded pole motors, in pressed steel frames, is available in two frame sizes, with capacities ranging from approximately 1/400 to 1/25 hp. Frames are of pressed steel and are equipped with self-aligning, wool-packed, oil impregnated bearings and surrounding oil retaining washer. Motor design includes a covered bearing in the back cover, which has a fibre thrust plug to absorb the thrust action of the shaft. The small frame is furnished with a detachable strap base and the large frame can be furnished with rigid saddle



base or resilient hub mounting. All ratings can be furnished without base, with extended through-bolts for stud mounting. Emerson Electric Mfg. Co., St. Louis, Mo.

Hammertone Finish

Finishes resembling hammered silver, copper, bronze, and other ornamental metals can be applied to products made of any kind of metal or molded plastics by the use of a new type of finish known as Hammertone. In finishing a surface, a base coat of the desired color is sprayed on first. This is immediately followed by a spatter coat of Hammertone liquid, producing the hammered pattern in the base coat. The product is then baked at medium heat for one hour, producing a lustrous metallic finish that is very tough and durable. Maas & Waldstein Co., 438 Riverside Ave., Newark, N. J.

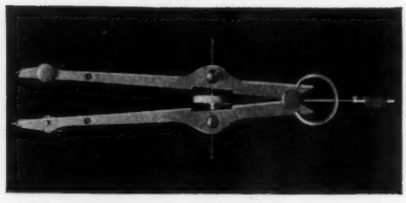
Gear Shift Motor

Automotive gear shift provides for instant interchangeability in speed without shifting any belt. It is built as an integral part of the motor and also as a gear shift drive which can be used

with any standard motor. The gear shift gives three shifts forward and one reverse, or four speeds forward. The complete motor and gear shift mechanism can be installed on lathes, milling machines, jointers, planers, drill presses, grinders and other machines to eliminate counter-shafts and the costly nuisances of central power drives. Lima Electric Motor Co., 440 N. Main St., Lima, Ohio.

Bow Pencil Compass

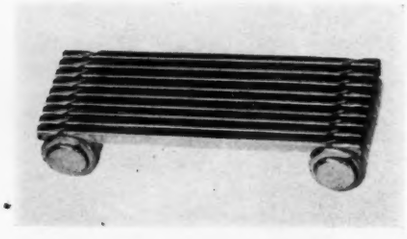
Open truss design of this bow pencil compass makes possible the construction of a large compass with a 40 per cent reduction in weight. Instrument is constructed of steel, except for the center screw thumb-piece and the handle, which are of Dural. The center screw



articulates with the legs through cylindrical nuts. Legs bear upon a double-grooved hinge pin of broad base, insuring strength and perfect alignment. Compass measures 6½ in. in length and will describe circles up to 10 in. in diameter. V. & E. Engineering Co., 758 South Fair Oaks Ave., Pasadena, Calif.

Wedge Wire

Wedge wire, wedge-shaped or undercut bars formed integral with solid heavy loops spaced at definite intervals, formerly available only from Europe, are now being made in this country. These "loop-bar" strands are securely fixed in grid form by strong cross rods which are firmly riveted or bolted in



place. Such construction gives dependable accuracy, strength and uniformity for high capacity sieve and screen applications. Can be made in practically any metal. Wedge Wire Corp., 5200 Harvard Ave., Cleveland, Ohio.

Manufacturers' Publications

Materials

MEEHANITE—Meehanite Research Institute of America, Inc., 311 Ross St., Pittsburgh, Pa. Bulletin 9, 16 pages, 8½x11 in. Sets forth engineering properties of Meehanite castings, and describes applications of the metal in machine tools.

NEOPRENE—Rubber Chemicals Div., E. I. du Pont de Nemours & Co., Inc., Wilmington, Del. Motion picture, with sound, 20 min. long. Shows how neoprene is made, explaining its chemical formula by means of animated cartoons. It also shows interesting tests of the material.

STAINLESS TUBING — J. Bishop & Co. Platinum Works, Stainless Steel Products Div., Malvern, Pa. Bulletin, 8 pages, 8½x11 in. General description of Bishop stainless steel seamless tubing. Mechanical, capillary, and hypodermic needle tubing are covered.

WIRE PRODUCTS—Allegheny Ludlum Steel Corp., 2319 Oliver Bldg., Pittsburgh, Pa. Blue Sheet, 8 pages, 8½x11 in. Gives complete specifications for all types of Allegheny metal wire products. Principal uses, analysis and types available are listed.

Z METAL—Ferrous Metals Corp., 444 Madison Ave., New York, N. Y. Catalog, 10 pages, 8½x11 in. Lists commercial applications, composition, physical, metallurgical and mechanical properties of Z Metal, a ductile alloy cast iron.

Mechanical Parts

BLOWER WHEELS — Advance Aluminum Castings Corp., 2742 W. 36th Pl., Chicago, Ill. Catalog, 80 pages, 9x7½ in. describes complete line of "Castalu" fan wheels, including many performance charts and tables showing the capacities of different size fans.

BRONZE BUSHINGS—Atlas Brass Foundry, Inc., Los Angeles, Calif. Catalog 30, 64 pages, 8½x11 in. Stock listings of bronze bushings, bearings, and rough and finished stock. Bearings made to specifications are also listed, as are many other Atlas bronze specialties.

COMPRESSION FITTINGS—Hays Mfg. Co., Erie, Pa. Bulletin Section 3, 28 pages, 8½x11 in. Sets forth features of Hays Double-Seal fittings for copper pipe. Results of extensive tests made on this type of fitting are reported, and available types are listed.

FLEXIBLE COUPLINGS — Ajax Flexible Coupling Co., Westfield, N. Y. Data Book, 20 pages, 8½x11 in. This handbook, planned for design engineers, gives data on horsepower, speeds and torque ratings, as well as dimensional specifications on forged steel and cast iron couplings.

PULLEYS—Boston Gear Works, Inc., Hancock & Hayward Sts., North Quincy, Mass. Bulletin, 8 pages, 8½x11 in. Illustrates, gives dimensions, diagrams and complete engineering data for pulleys, shaft supports, and couplings of the Hardware Products Co.

RUBBER MOUNTING—Lord Mfg. Co., 1639 W. 12th St., Erie, Pa. Bulletin 103, 8½x11 in. Describing "V-S" plate form bonded rubber mounting, this bulletin gives data as to dimensions, loads, styles and methods for choosing flexibility for a shock absorber.

V-BELT DRIVE—Allis-Chalmers Mfg. Co., Texrope Div., Milwaukee, Wis. Bulletin B-6051-A, 16 pages, 8½x11 in. Presents facts and data essential in estimating costs, types and sizes of V-belt drive equipment to buyers in every industry. Complete sheave information is included.

Electrical Parts

ELECTRICAL CONTACTS—Gibson Electric Co., 8350 Frankstown Ave., Pittsburgh (21), Pa. Catalog C-10, 16 pages, 8½x11 in. Comprehensive discussion of electrical contacts and contact materials. Features of Gibsiloy contacts are set forth, and many engineering diagrams show shapes available and mechanical applications of contacts.

HEATING UNITS — Watlow Electric & Mfg. Co., 1319 N. 23rd St., St Louis, Mo. Catalog, 22 pages, 9x11¼ in. Giving many illustrations of applications of Watlow units, this catalog lists complete line of cartridge, strip and immersion units and temperature controls for heating units.

Miscellaneous

ENGINE INDICATORS—Bacharach Industrial Instrument Co., 7000 Bennett St., Pittsburgh, Pa. Bulletin 279, 4 pages, 8x10½ in. Sets forth features of the Type 2-CP engine indicator for use on medium and moderately high speed engines. Bulletin 281, 4 pages, 8x10½ in. Type 4-HS engine indicator for indicating high speed engines.

LOCALIZED LIGHTING—Fostoria Pressed Steel Corp., Fostoria, Ohio. Catalog ML-25, 24 pages, 4x8½ in. Catalogs Fostoria lighting fixtures which are suitable for localized lighting, and illustrates numerous applications of each.

WRENCH CLEARANCES

T. B. JEFFERSON

THE SELF-EXPLANATORY TABLES given on these pages tell at a glance the size of the wrench-head, the clearance and spacing requirements, and fillet radii necessary for any given size of bolt up to 3 in. in diameter. These tables have been calculated from approved formulas, and have been found by experience to be of great help to design engineers.

All dimensions are based upon the newer American Standard Bolt and Nut dimensions, approved by the American Institute of Bolt, Nut and Rivet Manufacturers and adopted by the American Standards Association. As a result of the adoption of these newer standards, the American Standard Regular has replaced that series of bolts and nuts formerly known as "Manufacturers Standard," "American Budge Standard," etc., while the "U. S. Standard"

Table I—Open End Wrenches; Dimensions, Clearances and Bolt Spacing*

Bolt Size A	Nut Dimensions		Wrench Radii			Corner Clearance	Row Clearance	Bolt Spacing
	Short Dia. $B = 1\,1/2A + 1/8$	Long Radius $C = 0.577B$	$R_1 = 0.6682B$	$R_2 = 0.759B$	$R_3 = 1.3365B$	$X = 1.1682B$	$Y = A + 1/8$	$Z = 1.7457B + 0.5A$
1/4	1/2	9/32	21/64	3/8	21/32	37/64	3/8	63/64
3/8	11/16	25/64	29/64	33/64	29/32	51/64	1/2	1 3/8
1/2	7/8	1/2	37/64	21/32	1 5/32	1 1/64	5/8	1 49/64
5/8	1 1/64	39/64	45/64	51/64	1 13/32	1 15/64	3/4	2 5/32
3/4	1 1/4	23/32	53/64	15/16	1 21/32	1 29/64	7/8	2 35/64
7/8	1 7/16	53/64	61/64	1 5/64	1 29/32	1 43/64	1	2 15/16
1	1 5/8	15/16	1 5/64	1 7/32	2 5/32	1 57/64	1 1/8	3 21/64
1 1/8	1 13/16	1 3/64	1 13/64	1 23/64	2 13/32	2 7/64	1 1/4	3 23/32
1 1/4	2	1 5/32	1 21/64	1 1/2	2 21/32	2 21/64	1 3/8	4 7/64
1 3/8	2 3/16	1 17/64	1 29/64	1 41/64	2 29/32	2 35/64	1 1/2	4 1/2
1 1/2	2 3/8	1 3/8	1 37/64	1 25/32	3 5/32	2 49/64	1 5/8	4 57/64
1 5/8	2 9/16	1 31/64	1 45/64	1 59/64	3 13/32	2 63/64	1 3/4	5 9/32
1 3/4	2 3/4	1 19/32	1 27/32	2 3/32	3 11/16	3 13/64	1 7/8	5 43/64
1 7/8	2 15/16	1 11/16	1 61/64	2 7/32	3 29/32	3 27/64	2	6 1/16
2	3 1/8	1 13/16	2 3/32	2 3/8	4 3/16	3 21/32	2 1/8	6 29/64
2 1/8	3 5/16	1 59/64	2 7/32	2 33/64	4 27/64	3 7/8	2 1/4	6 27/32
2 1/4	3 1/2	2 1/32	2 11/32	2 21/32	4 11/16	4 3/32	2 3/8	7 15/64
2 1/2	3 7/8	2 15/64	2 37/64	2 59/64	5 5/32	4 33/64	2 5/8	8 1/64
2 3/4	4 1/4	2 29/64	2 53/64	3 11/32	5 21/32	4 61/64	2 7/8	8 51/64
3	4 5/8	2 43/64	3 5/64	3 31/64	6 5/32	5 25/64	3 1/8	9 9/16

* All dimensions given in this table are expressed in inches

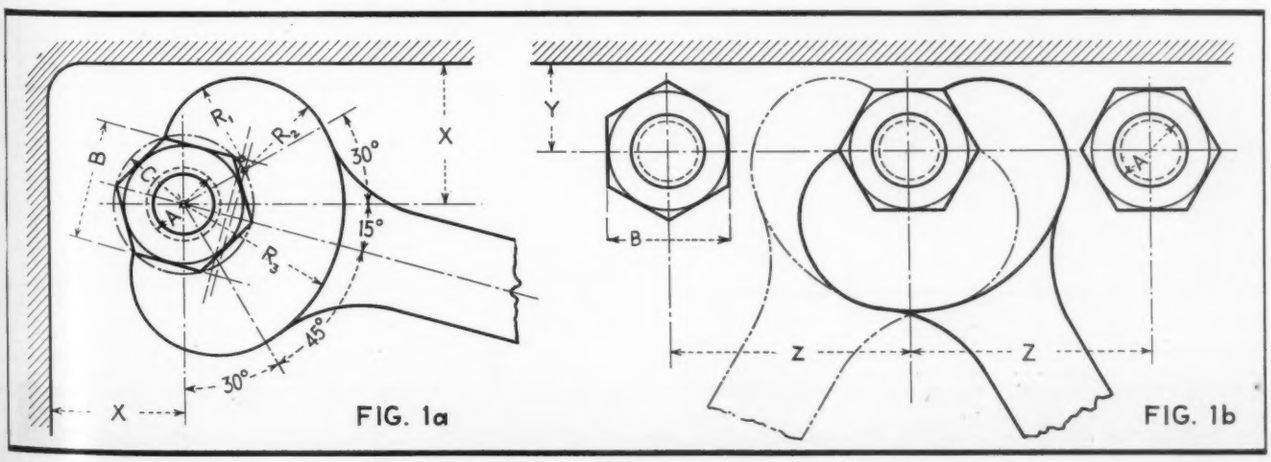


FIG. 1a FIG. 1b

WRENCH CLEARANCES *(Continued)*

series have been replaced by the American Standard Heavy.

Since some wrench manufacturers prefer to make small wrench sizes with American Standard Heavy wrench dimensions, all dimensions given in this article are based upon the short diameter of the nut for the American Standard Heavy series. In practice bolt heads are made to American Standard Regular dimensions. Nut dimensions for bolts up to 1⅜ in. are American Standard Regular, ⅓ in. and 1½ in. nuts may be either American Standard Regular or Heavy, and for bolt diameters over 1½ in. the nuts are dimensioned in accordance with the American Standard Heavy series.

The data given in this article is sufficient to solve almost any bolt spacing problem, without resorting to the application of formulas. In general, it should be noted that the use of socket wrenches permits closer bolt spacing, but necessitates the specification of larger wall clearances than is called for by the use of open end wrenches.

Table II—Socket Wrenches; Wrench Clearance and Bolt Spacing

Bolt Size *A* in.	Wrench Diameter *D* in. (See Note a)	Edge Distance $E = A + 1/8$	Clearance *F* in. (See Note b)	Bolt Spacing $G = 2A + 1/4$
1/4	3/4	3/8	9/16	3/4
3/8	1 1/8	1/2	3/4	1
1/2	1 1/2	5/8	15/16	1 1/4
5/8	1 3/4	3/4	1	1 1/2
3/4	2 3/32	7/8	1 5/32	1 3/4
7/8	2 29/64	1	1 21/64	2
1	2 19/32	1 1/8	1 11/32	2 1/4
1 1/8	2 59/64	1 1/4	1 1/2	2 1/2
1 1/4	3	1 3/8	1 41/64	2 3/4
1 3/8	3 19/64	1 1/2	1 51/64	3
1 1/2	3 39/64	1 5/8	1 61/64	3 1/4
1 5/8	3 29/32	1 3/4	2 7/64	3 1/2
1 3/4	4 13/64	1 7/8	2 1/4	3 3/4
1 7/8	4 1/2	2	2 13/32	4
2	4 51/64	2 1/8	2 35/64	4 1/4
2 1/8	5 3/32	2 1/4	2 45/64	4 1/2
2 1/4	5 13/32	2 3/8	2 55/64	4 3/4
2 1/2	5 45/64	2 5/8	3 5/32	5 1/4
2 3/4	6 39/64	2 7/8	3 29/64	5 3/4
3	7 13/64	3 1/8	3 3/4	6 1/4

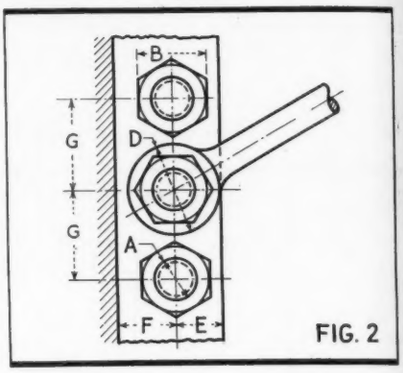


FIG. 2

Note (a)

$D = 3.0\ A$ for bolts 1/2″ in dia. or less
$D = 2.8\ A$ for bolts from 5/8″ to 7/8″
$D = 2.6\ A$ for bolts 1″ to 1 1/8″ in dia.
$D = 2.4\ A$ for bolts from 1 1/4″ to 3″
$D = 2.3\ A$ for bolts above 3″ in dia.

Note (b)

$F = 1.5\ E$ for bolts 1/2″ in dia. or less.
$F = 1.33\ E$ for bolts from 5/8″ to 7/8″
$F = 1.2\ E$ for bolts above 1″ in dia.

Table III—Fillet Radii

Bolt Size *A* in.	Fillet Radius *r* in.	Bolt Size *A* in.	Fillet Radius *r* in
1/4	1/8	1 1/2	5/8
3/8	3/16	1 5/8	11/16
1/2	1/4	1 3/4	11/16
5/8	5/16	1 7/8	11/16
3/4	3/8	2	11/16
7/8	7/16	2 1/8	11/16
1	7/16	2 1/4	3/4
1 1/8	1/2	2 1/2	13/16
1 1/4	9/16	2 3/4	15/16
1 3/8	5/8	3	15/16

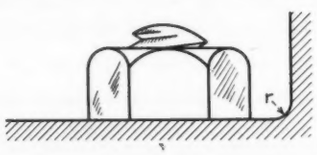


FIG. 3

Note—Formulas from which these radii are calculated are:

$r = 0.50\ A$ for bolts 7/8″ in dia. or less.
$r = 0.45\ A$ for bolts from 1 1/8″ to 1 3/8″ in dia.
$r = 0.42\ A$ for bolts from 1 5/8″ to 1 3/4″ in dia.
$r = 0.33\ A$ for bolts from 2 1/4″ to 2 3/4″ in dia.
$r = 1\ 1/8$″ for bolts from 3 1/4″ and above in dia.

For bolt sizes not included above use *r* for next smallest size bolt.

Ford Motor Company

AUTOMOBILES OF 1941

AUTOMOTIVE ENGINEERS have made many significant changes in the new cars with reference to improved engine performance, stronger frames, improved rubber mountings and new automatic transmissions. Among the new outstanding developments is the new Nash 600, an all welded car designed on the aircraft principle of using the body as a load-carrying member. With this type of construction, Nash engineers were able to reduce dead weight 500 lb. and produce a standard-size, high-economy car which could be sold in the low-price range. Chrysler engineers have adopted 0.004 in. babbitt bearings throughout their entire line. Buick has an extremely thin babbitt bearing, with babbitt thickness of 0.002 in. Bearing has a powder metal matrix.

Hydraulic couplings and automatic transmissions are being used more widely. The hydraulic coupling, last year available on the Chrysler Imperial, is now optional equipment on all Chrysler, Dodge and DeSoto cars. A semi-automatic, four-speed transmission which operates in conjunction with a hydraulic coupling, has been designed for the Chrysler six and DeSoto.

"Torpedo" styling is found on almost every line; in some cases, to accommodate the wider bodies, rear treads have been increased up to 2½ in. General Motors cars are using considerable more chromium trim this year, and running boards on most cars are concealed.

Increased use of plastic is noted, particularly with reference to injection-molded cellulose acetate. The light-carrying properties of acrylics are used to illuminate the speedometer needle of Packard and Chrysler cars, and the Hudson instrument panel. Extruded plastic trim is found on Ford, Mercury and Zephyr.

Additional horsepower and torque have been obtained without increase in size by refinements in engine design. For instance, combustion chamber and piston head contours have been changed to obtain higher compression of the new high-octane fuels. Buick has developed an interesting carburetion set-up, utilizing two dual carburetors and dual manifolds. Called compound carburetion, the scheme allows the engine to operate on one carburetor during medium-speed and low-speed driving. For wide-open speed and quick acceleration, the linkage is designed so that the second carburetor automatically cuts in. With this device, engine horsepower can be stepped up from 115 to 125, with gasoline consumption considerably decreased.

Chrysler AUTOMATIC DRIVE

Vacuum-Operated, Four-Speed Transmission Combined with Fluid Coupling

DRIVING EASE, together with reduction of transmitted torsional vibrations and shocks and increased life in driven parts, has led to more and more widespread public acceptance of the fluid coupling in the automotive field. Operating on a fixed one-to-one torque ratio and mounted directly on the engine crankshaft, this coupling was first used in motor cars on the English Daimler in 1928. Known as the fluid drive, it was first offered in this country by Chrysler in its 1939 Imperial line. The notable success and wide public acceptance of the device both from the engineering and customer angles has led to its adoption on 1941 Chrysler, DeSoto and Dodge models.

The history of the hydro-kinetic power transmitter dates back to the early 1900's when Dr. Föttinger designed for the Vulcan Company a hydraulic torque transformer which was used between a marine steam turbine and propeller. It had five-to-one reduction and an efficiency of 85 per cent. With the development of less expensive, more efficient helical-gear reducers this torque transformer was displaced. Dr. Bauer of the same company later redesigned it, producing a two-element transmitter operating on the Föttinger principle and furnishing one-to-one torque ratio. It was this coupling that Harold Sinclair of Isleworth, England, modified in 1928 for use with internal combustion engines.

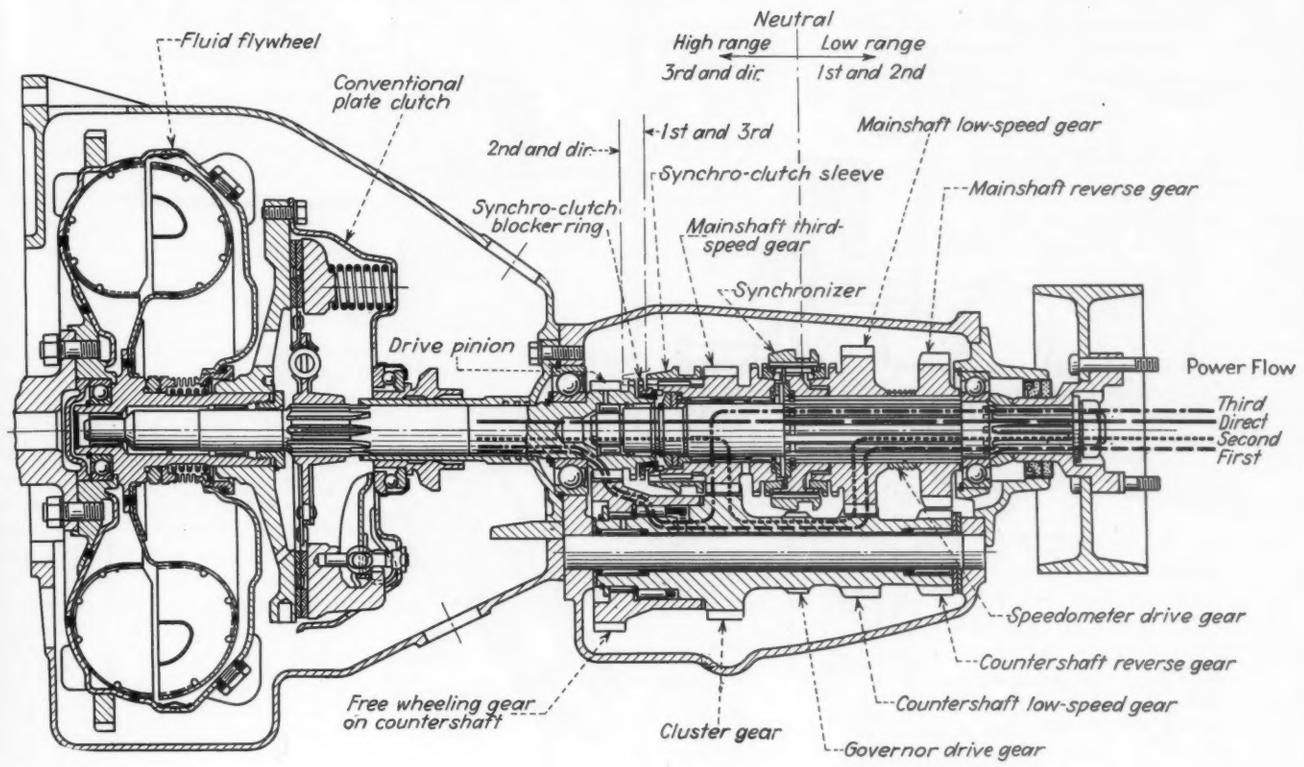
A conventional manually operated clutch is usually retained with the fluid coupling plus an auxiliary transmission. Although the coupling permits almost complete operation of the car in high gear, the clutch is provided for selection of first and second for starting on hills, for quick acceleration and for reverse.

Early experience with fluid drive and with the development of automatic shifting on transmissions led Chrysler Corporation to its latest development for 1941. This new transmission represents a step in the gradual evolution from the old three-speed gear box with its "wiggle stick" through the automatic overdrive and remote control gear shift stage to its latest development. During this evolution, however, Chrysler Corporation engineers have presented successive models of transmissions in such a way that the driver of new models found himself able to handle each new transmission with ease and without acquiring a complete new set of driving habits in order to control his automobile with the minimum of effort.

Selection of the proper gear ratio for driving conditions is done automatically in the new Chrysler Six Vacamatic and DeSoto Simplimatic drive, which has four forward gear ratios: in the "low" range, 3.07 to 1, and 1.98 to 1 ratios; in the "high" range, 1.55 to 1, and 1 to 1 ratios. The proper ratio in the manually selected range is automatically chosen by a vacuum-operated linkage electrically interlocked with the accelerator and transmission governor. The Oldsmobile Hydramatic transmission, which was described in P.E. *Nov. 1939, pages 466–469*, utilizes a fluid coupling operating in conjunction with a planetary transmission. Hydraulic cylinders and a governor automatically select one of four gear ratios according to speed and load.

The new Chrysler underdrive transmission, in conjunction with a fluid drive unit, allows the driver to do 98 percent of his driving with the shift lever in the "high" position, without the necessity of depressing the clutch pedal or of shifting the gearshift lever. There is no need to shift gears or tire the left foot with excessive clutch pedal operation. When the car reaches a speed of between 15 and 17 mi. per hr. or higher, momentary release of foot pressure on the accelerator pedal allows the transmission to shift automatically into cruising gear, in which engine speed is reduced. Depression of the accelerator pedal past the wide-open position causes the transmission to shift automatically into a lower gear; it will again shift automatically into cruising gear when the accelerator is momentarily released.

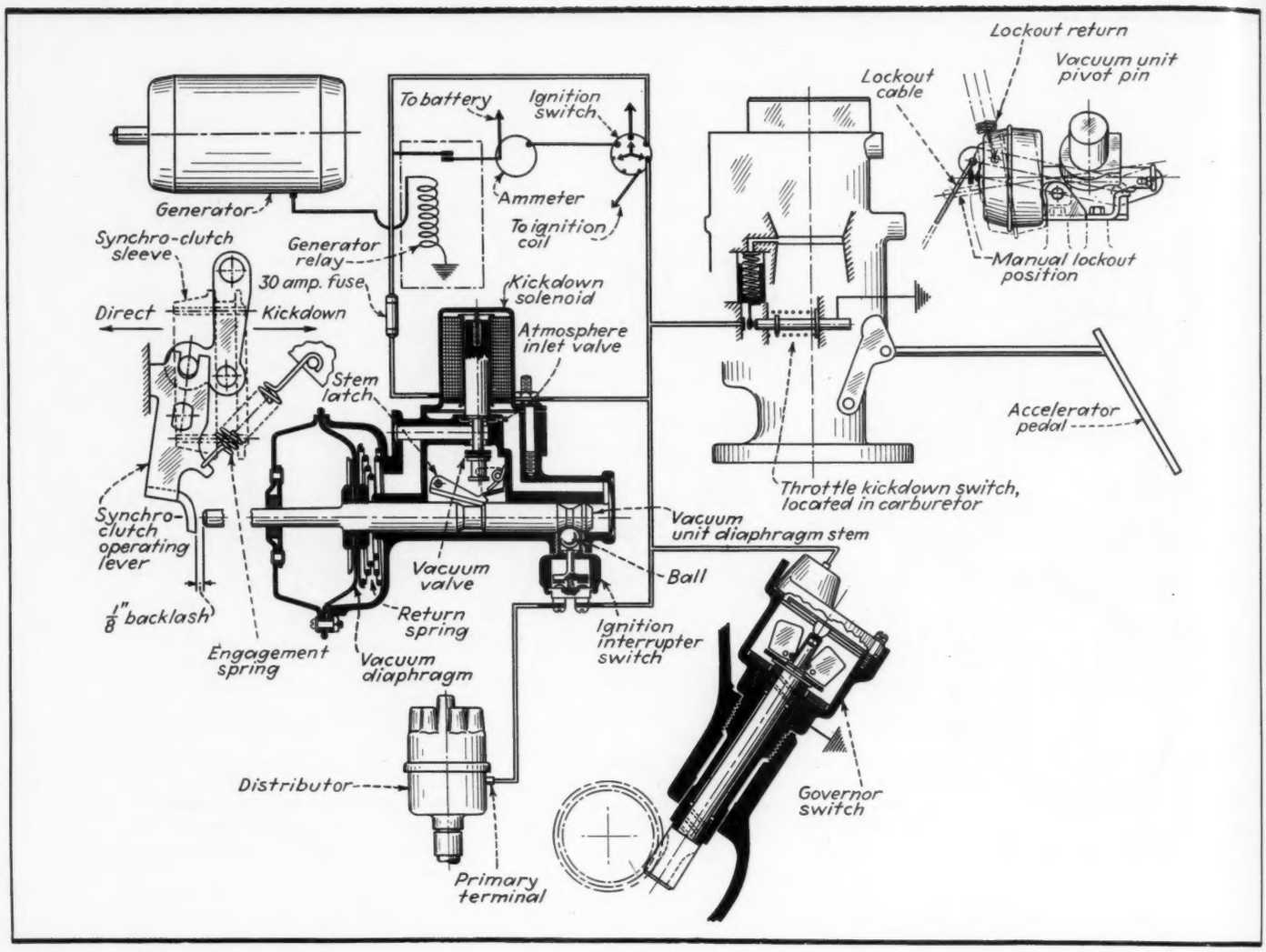
For climbing steep hills and for driving in sand, gravel, and rough terrain, "low range" is selected by moving the shifting lever manually to the position normally occupied by second speed in the conventional transmission. In this "low range", the car is in first speed when starting and shifts automatically to second speed as the accelerator pedal pressure is momentarily relaxed with car speed above approximately 8 mi. per hr. A "step-down" feature allows the transmission to be shifted automatically from second to first speeds by pressing the accelerator pedal past the wide open throttle position. The action in "stepping down" to a lower gear with the underdrive transmission is exactly the same as in a car equipped with overdrive transmission, insofar as the driver controls are concerned. Reverse is selected as with a conventional transmission. With this transmission two reverse ratios are available.



Cross-sectional view of the fluid drive unit, clutch, clutch housing, and underdrive transmission. The fluid drive unit is bolted directly to the crankshaft, and a conventional type plate clutch is attached to the output member of the unit; this in turn rotates the drive pinion of the transmission which meshes with the free-wheeling gear on the countershaft. This gear drives, through free-wheeling unit, the cluster gear, the countershaft third speed gear, the countershaft low speed gear, and the countershaft reverse gear. The mainshaft third speed gear, mounted on the mainshaft on roller bearings, may be directly clutched to the drive pinion by means of a synchro-clutch sleeve. The mainshaft third speed gear may be directly coupled to the mainshaft by sliding the sleeve of synchronizer forward into engagement with the clutching teeth of the mainshaft third speed gear. The hub of manual synchronizer is splined to the mainshaft, and this synchronizer serves also to couple the mainshaft low speed gear to the mainshaft when the sliding sleeve is moved mackward so as to engage the clutching teeth on mainshaft low speed gear. Mainshaft reverse gear is splined to the mainshaft. Reverse idler gear, mounted on an individual shaft, may be manually moved into mesh with the countershaft reverse gear and the mainshaft reverse gear. Speedometer-drive gear is splined to the mainshaft and meshes with a speedometer drive pinion mounted in a retainer bolted to the transmission case. Governor drive gear, cut integral with the cluster gear, meshes with the governor drive pinion.

The synchro-clutch, which engages when an automatic shift is made from first to second, or from third to fourth speeds, couples drive pinion with mainshaft third speed gear. Clutch sleeve is splined to mainshaft third speed gear. Synchro-clutch blocker ring is mounted between mainshaft third speed gear and drive pinion in such a manner as to prevent the forward movement of synchro-clutch sleeve into engagement with clutching teeth on the rear end of drive pinion until the two parts have come to the same speed of rotation. In order to accomplish this "blocking" action, and prevent the engagement of the sleeve with the drive pinion clutching teeth, the blocker ring rotates a sufficient amount, with respect to third speed gear, to cause the ends of the teeth on sleeve to butt with the teeth on the outside of blocker ring at all times when the drive pinion and the mainshaft third speed gear are not rotating at the same speed. Synchronizer is type used on 1940 Chrysler cars.

(*Continued on next page*)



Control system of the underdrive transmission engages and disengages the synchro-clutch through the action of mechanical, electrical, and vacuum units. By means of this system the transmission in the "low range" shifts automatically from first to second speeds at car speeds above approximately 8 mi. per hr. when the driver momentarily lifts his foot from the accelerator pedal; shifts automatically from second to first speeds when the car speed drops below approximately 7 mi. per hr.; shifts from second to first speeds when the driver depresses the accelerator pedal past the wide open position at car speeds below approximately 27 mi. per hr.; in the "high range" shifts automatically from third to fourth speeds at car speeds above 15 to 17 mi. per hr. when the driver momentarily lifts his foot from the accelerator pedal; shifts automatically from fourth to third speeds when the car speed drops below 13 to 15 mi. per hr.; shifts from fourth to third when the driver depresses the accelerator pedal past the wide open position at car speeds below approximately 53 mi. per hr.

The control system vacuum unit assembly is mounted on the right side of the transmission. It contains the kick-down solenoid and the ignition interrupter switch. The governor assembly, mounted in the right side of the transmission case, contains the governor switch. The throttle kick-down switch, located in the carburetor; the governor relay, the lockout mechanism, comprising cable lever, and return spring, are remaining parts.

The transmission synchro-clutch, controlling the shifts from first to second and from third to fourth speeds, is engaged through the action of engagement spring on synchro-clutch operating lever when vacuum unit diaphragm stem latches and moves out of the way of operating lever, or when the manual lockout cable is pulled out, thus rotating vacuum unit about its pivot pin and allowing synchro-clutch operating lever to move independently of the vacuum unit diaphragm stem. The vacuum unit diaphragm stem is shown in the latched position, accomplished through the action of engine manifold vacuum on the right side of vacuum diaphragm. As the stem moves to the right, the return spring is cocked, and when the stem has moved to the right into the position shown in the diagram, the stem latch drops into latching groove in the stem. The chamber behind the vacuum diaphragm is connected to the engine manifold vacuum through a vacuum valve mounted on the stem of the solenoid core. This valve is open to admit vacuum to the diaphragm when the solenoid is not energized and when the core is in the position shown. The downshift or kick-down of the synchro-clutch is accomplished by energizing solenoid to raise solenoid core, thus opening atmosphere inlet valve and closing vacuum valve, admitting atmospheric pressure behind vacuum diaphragm. As the solenoid core moves upward, it pulls diaphragm stem latch out of the groove in the diaphragm stem, thus allowing the previously cocked return spring to move the diaphragm stem to the left. As the dia-

phragm stem moves to the left, ignition interrupter switch ball is forced out of the groove in the diaphragm stem, closing ignition interrupter switch, which interrupts the ignition. As the ignition is interrupted, the car moves from drive to coast, at which instant the synchro-clutch sleeve is free to move on its hub, thus allowing the vacuum unit diaphragm stem to move all the way to the left, at the same time moving synchro-clutch operating lever to the disengaged position. When the vacuum unit diaphragm stem has moved all the way into the unlatched position, interrupter switch ball is again free to move inward under the force of the switch spring, thus opening the ignition interrupter switch, and restoring ignition.

One terminal of solenoid is connected to the battery through the ignition switch and through generator relay, and is "hot" at all times when the engine ignition switch is turned on, and when the generator is rotating above a minimum speed. The other terminal of the solenoid is the ground side, and thus may be grounded through the governor switch, through the kick-down switch, or through the ignition switch when it is turned to "off" position.

The governor which controls the car speed at which a shift may be made from first to second speeds in the "low range" and from third to fourth speeds in the "high range," is driven from the transmission cluster gear, and inasmuch as the cluster gear rotates roughly twice as fast in first and second speed as it does in third and fourth speeds for an equivalent car speed, the cut-in and cut-out speeds of the governor in the "low range" will be approximately half of the cut-in and cut-out car speeds in the "high range." Outward movement of the governor weights moves the switch plunger upward and causes the switch to open at speeds above the cut-in speed, and to close at speeds below the cut-out speed. With the governor switch closed, the solenoid is grounded and energized, causing the transmission to stay in first or third speeds. When the governor switch is opened at the cut-in speed of the governor, the ground circuit to the solenoid is broken, and the vacuum unit stem latches, making it possible for the synchro-clutch to engage when foot pressure is released from the accelerator pedal.

The kick-down switch is closed whenever the throttle is moved past the wide open position. Closing of this switch grounds the solenoid, causing the transmission to be shifted into third or first speeds. When the throttle is again partly closed, this throttle switch opens and allows the vacuum unit stem to latch, providing for the engagement of the synchro-clutch when foot pressure is removed from the accelerator. This switch is prevented from closing above a certain engine speed by the action of the carburetor venturi vacuum on a small plunger, thus making it impossible to cause shifting by a kick-down on the accelerator pedal at speeds above approximately 27 mi. per hr. in the "low range" and above approximately 53 mi. per hr. in the "high range."

The generator relay and the special ignition switch with a grounded terminal in the "off" position insure that the transmission will always start in first or third speeds, even though the ignition is turned off with the car still moving. With the ignition switch turned off, the solenoid ground connection is established through the grounded terminal of the ignition switch, and the "hot" lead is maintained through the action of a generator relay, which closes the circuit from the solenoid to the battery whenever the generator is rotating at a speed above a small minimum. When the engine and generator stop rotating, generator relay opens, the "hot" lead to the solenoid is broken, and the solenoid is de-energized.

Power flow, 1st speed. The manual shift lever is shifted from neutral to the position normally occupied by second speed in the conventional transmission. This moves manual synchronizer sleeve backward, locking mainshaft low speed gear to mainshaft. Power then flows from drive pinion into free-wheeling gear on the countershaft, through locked free-wheeling unit into countershaft low speed gear and thence into mainshaft low speed gear, which turns the mainshaft through manual synchronizer sleeve. The transmission free-wheels when coasting in first speed. Ratio 3.07:1.

2nd speed. Manual shift lever in second speed remains in the same position as for first speed. Shift from first to second speeds is through the action of governor in causing vacuum unit to latch, thus permitting synchro-clutch sleeve to engage the clutching teeth of the drive pinion. Power then flows from drive pinion through synchro-clutch sleeve into mainshaft third speed gear, down into countershaft third speed gear, through countershaft low speed gear and into mainshaft low speed gear, which turns the mainshaft through manual synchronizer sleeve. The transmission does not free-wheel in second speed. Ratio 1.98:1.

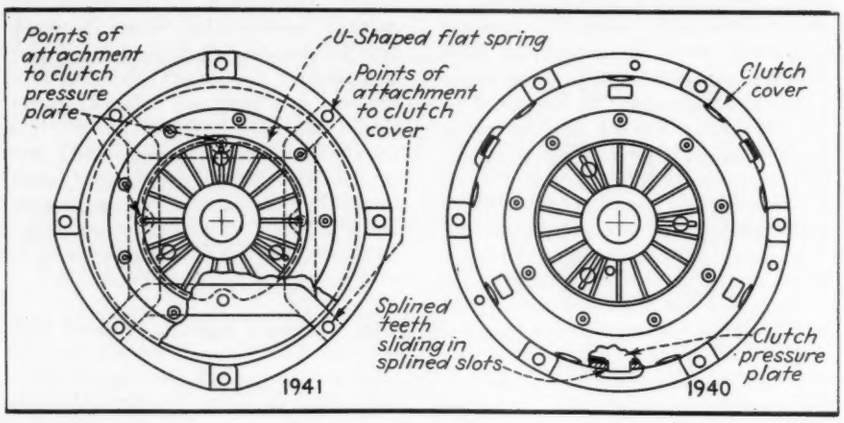
3rd speed. Manual shift lever is shifted into the position normally occupied by the high speed in the conventional transmission. This moves the manual synchronizer sleeve forward, thus locking mainshaft third speed gear to mainshaft. Synchro-clutch sleeve is in the rear or disengaged position. Power then flows from drive pinion into countershaft free-wheeling unit, into countershaft third speed gear, and thence into mainshaft third speed gear and into mainshaft through manual synchronizer sleeve. The transmission free-wheels in third speed. Ratio 1.55:1.

4th (Direct) speed. Manual shift lever in fourth speed remains in the same position as for third speed. Shift from third to fourth speeds is through the action of governor in causing vacuum unit to latch, thus permitting synchro-clutch sleeve to engage the clutching teeth of the drive pinion, and locking mainshaft third speed gear to the drive pinion. Power then flows directly from the drive pinion into the mainshaft through the two sleeves and third speed gear. The transmission does not free-wheel in fourth speed. Ratio 1:1.

Reverse speed. Manual shift lever is moved to the same position as occupied by the reverse speed in the conventional transmission. This moves reverse idler gear backward into mesh with countershaft reverse gear and mainshaft reverse gear. Power then flows from drive pinion into countershaft free-wheeling gear, through locked free-wheeling unit into countershaft reverse gear, and thence into reverse idler gear and into mainshaft reverse gear, which is splined to the mainshaft. The transmission free-wheels when coasting in reverse speed. Ratio 3.25:1.

Second reverse ratio may be obtained above the governor cut-in speed by momentarily releasing the foot from the accelerator and allowing the synchro-clutch to engage. Power then flows from drive pinion through synchro-clutch sleeve into mainshaft third speed gear, into countershaft third speed gear, into countershaft reverse gear, and thence through reverse idler gear into mainshaft reverse gear, which is splined to the mainshaft. The transmission does not free-wheel in this second reverse speed. Ratio 2.10:1.

Shifts from neutral into the "low range" (first and second speeds) and into the "high range" (third and fourth speeds) are made through a synchronizer, thus preventing clashing of gears. The shift from the "high range" to the "low range" may be made at any car speed by declutching and moving the manual shift lever from one position to the other. The clutch pedal must be depressed whenever shift lever is moved.



Spring-centered clutch pressure plate. Commonly, the pressure plate, which in its operation must move axially, is held concentric by three splined teeth cut on the rim of the plate which fit splined slots milled in the clutch cover. Since concentricity within close limits is required, spline fits must be close, but with enough clearance had to allow easy sliding, making a difficult production problem. In the 1941 Pontiac, four pieces of flat spring steel are blanked out of thin stock to a wide shallow U-shaped profile. Each end of the spring is riveted at one of four equally spaced holes in the clutch cover flange, the four springs being symmetrical with the clutch gear shift. Four rivets pass through holes drilled in the center of each spring and through four equally spaced lugs on the pressure plate. Thus the plate is held concentric but is free to move fore and aft when clutch is being engaged or released, the springs bowing slightly when the clutch is being engaged. The spring action assists the main disk spring to release plate when clutch is disengaged.

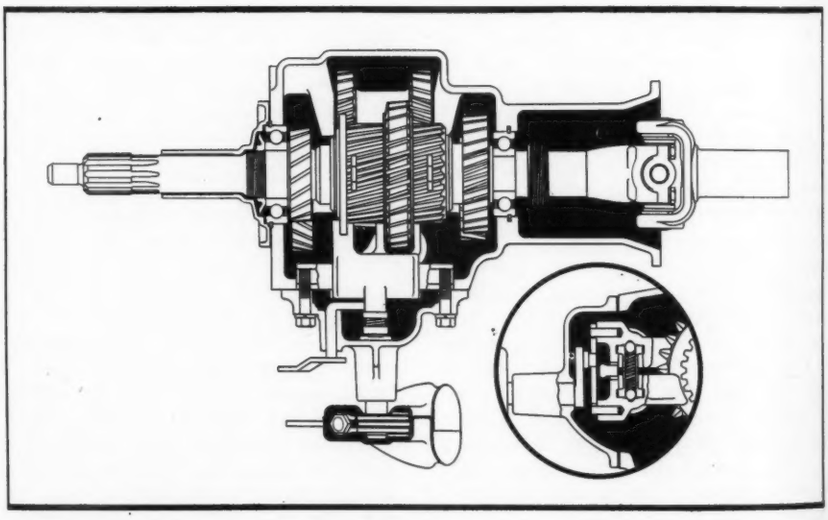
ALUMINUM BRONZE PISTON PIN bushings are used in the 1941 Pontiac to avoid possible corrosion under certain conditions of oil and heat. The formerly used bronze bushing was subjected to this corrosion in certain sections of the country. Although the failures have been comparatively few, the aim in specifying this type bushing for all units is to maintain 100 per cent performance and reduce complaints.

POWER OUTPUT in the Chevrolet engine is increased to 90 hp. with a maximum torque of 174 lb.ft. primarily through redesign of the combustion chamber and piston. Change in the combustion chamber resulted in raising the compression ratio from 6.25 to 1 to 6.5 to 1. This was accomplished by lowering the roof around the intake valve seat to the face of the cylinder head. The lower roof section is machined flush with the base. Likewise, the valve seat is lower and, to allow for the thickness of the valve head, is set in a shallow recess in the roof. When the piston is at the top of the stroke, the distance from it to the lowered roof area is only the thickness of the head gasket. This restricted volume increases the turbulence under compression, and permits operation on a leaner mixture. Another important result is the quenching of the flame over the last third of travel, to eliminate one of the major causes of detonation.

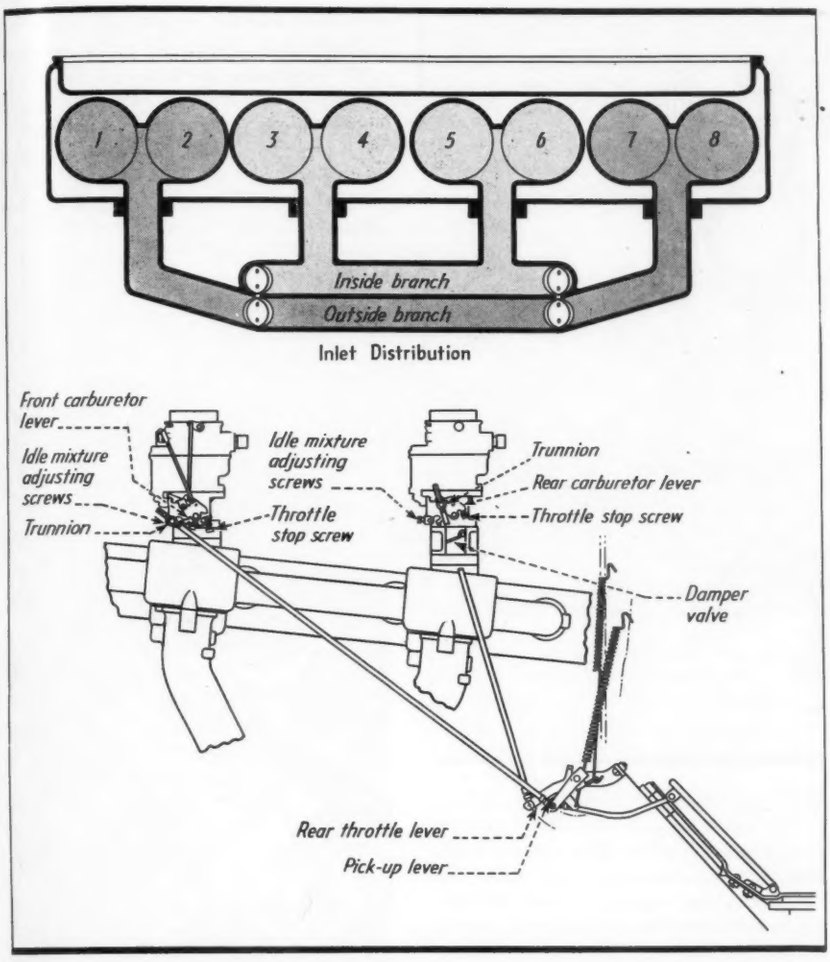
PACKARD CYLINDER BLOCKS for all engines are semi-steel castings containing a definite percentage of nickel. The exceptional hardness of this material makes unnecessary steel valve seat inserts.

TO PROTECT BREAKER POINTS in the 1941 Chevrolet, a polarity reversing switch is installed in the distributor circuit. Mounted on the top of the starter and actuated by the starting motor, the switch reverses the direction of current flowing to the breaker point each time the motor is started. By periodically reversing current direction, transfer of tungsten from one point to the other is also reversed, resulting in a more uniform condition of the points, and prolonging life. The breaker side on the distributor cam lobe is made steeper. This increases coil output at low speed and increases the operating speed of the breaker point to minimize any tendency of arcing at low engine speed. The cam is hardened by a special electrical process to permit reaming of the hole that supports the cam and the shaft, and to permit more accurate grinding of the cam.

CUSHIONING OF CLUTCH FACING in some form is universally used. In the 1940 Pontiac, cushioning was supplied by waved flat springs, which spread the two facings a few thousandths apart. For the 1941 models, springs have been omitted. In their place a portion of the steel disk underneath the clutch facing has been blanked into seven, equally spaced "T" sectors, wave-shaped. To the front face is riveted the front lining at the convex crest of each wave formation; rear facing is riveted to the centers of concave portion of each wave.



Mainshaft in the 1941 Chevrolet transmission now has six splines instead of 18. Lands between the splines are ground for accuracy of centering of the second and third speed clutch, the inside of which also is ground for the same purpose. Wear resistance of the shifter yokes is increased by Granodizing. Reverse idler gear bushings are locked into the gears. The bosses on the ends of the gear are enlarged for greater area of thrust washer contact. The interlocked lever is now clamped instead of welded to the interlock shaft; this permits removal of parts from the gearshift cover when service is required.



Compound carburetion is standard equipment on all Buick engines except the Series 40, and at small extra cost on that series. In the past with one carburetor supplying fuel for both high and low speeds as well as acceleration and hill climbing, a compromise had to be made between performance and economy. Two dual carburetors are used; each slightly smaller than in previous models. Front carburetor alone will supply a sufficient charge of fuel to all eight cylinders because of its fuel construction under ordinary driving conditions. Equipment includes a dual system of intake and to exhaust manifolds and carburetor heat controls. The automatic choke system and starter switch are continued on the front carburetor only. The front carburetor only has an accelerating pump system which gives a momentarily richer mixture when acceleration is depressed; and a main "by-pass" system which functions when throttles are nearly wide open.

On closed throttle or idling, air passing the front and rear throttles draws gasoline from the idle systems of both carburetors. With throttle partly open, gasoline is fed from the idle system of both carburetors and the main metering system of the front carburetor. For full throttle operation, usually in high gear, operation varies as follows: Below 15 m.p.h.—front carburetor main system is functioning—the rear carburetor is not; 15-20 m.p.h.—front carburetor main system is functioning and rear carburetor starts to feed gasoline mixture; 35-40 m.p.h.—both carburetor main systems function with accelerator.

Between the rear carburetor and the manifolds is an auxiliary throttle held closed by a weight and opened automatically by engine vacuum. Rear carburetor delayed action is accomplished by special linkage; it is not directly connected to the accelerator but is actuated by a pick-up lever with a delayed initial opening, as compared to the front carburetor throttle flies which are directly connected to the foot accelerator. When the front throttle is opened half way, the rear throttle begins to open.

Both carburetors are the concentric float bowl or aero type to assure a constant supply of fuel regardless of car elevation. Intake manifolds are designed to permit uninterrupted fuel passage to the cylinder. Exhaust manifolds are equipped with two automatic heat control valves, thermostatically controlled.

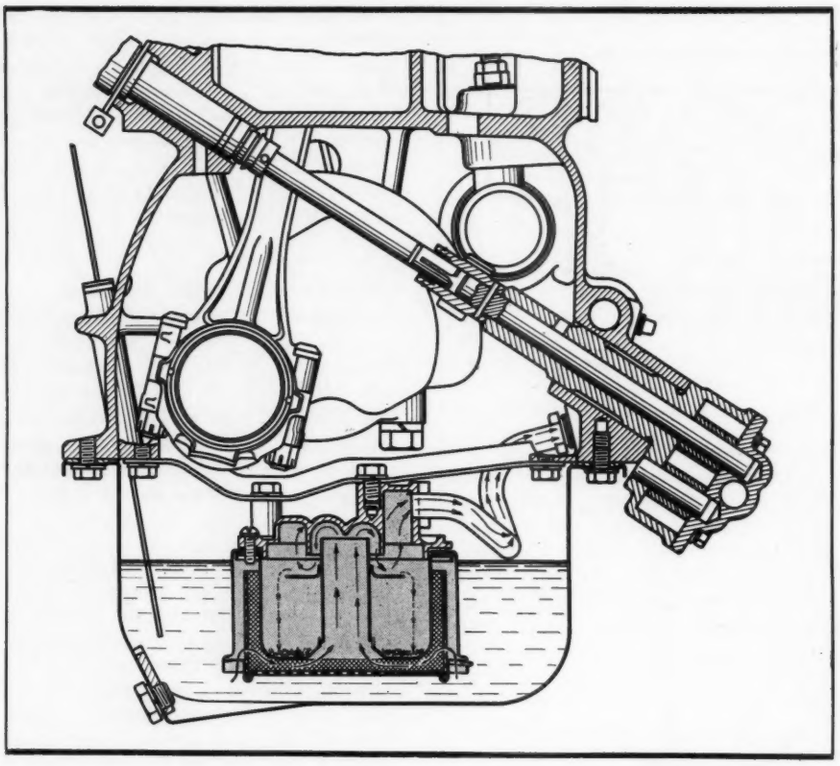
PRESSURE COOLING, first used on the 1940 Buick, is now on almost all 1941 cars. The systems usually operate under 4 lb. per sq. in. pressure. By the use of a radiator pressure cap the boiling points of the cooling mixture are raised from 10 to 12 deg. above normal, thus providing an extra margin of protection for the cooling systems.

WRIST PIN HOLES in the pistons are bearingized to improve Cadillac wrist pin fits. After the hole has been diamond-bored to size and anodized, the aluminum oxide formed by the anodizing process is cold-worked into the wall by the action of a many-toothed cam. This process lengthens the life of the anodizing and gives a mirror finish.

ALL PACKARD TRANSMISSION GEARS are carburized to impart a hard, high-carbon wearing surface on the outside of the gears while retaining sufficient interior toughness to withstand shock and impact. Ordinary heat-treating processes impart the same degree of hardness all the way through and gears properly hardened on the surface would be too hard and brittle on the inside. After carburizing, gears are lapped to precision limits and matched in sets.

INTAKE VALVES are 1/8th inch larger on the Packard and are made of chrome-nickel steel. Exhaust valves are made entirely from austenitic steel, this latter material is used in the heads to prevent warping and distortion under the extreme heat of the exhaust gases. The valve stems also are now made from austenitic steel to prevent corrosion. Valve stems operate in valve guides that are counterbored to shield the stem from hot gases. They are burnished on the inside to provide a smooth surface. In all engines the floor of the exhaust port has been raised to make a streamlined passage for the exhaust gases and also to provide more cooling around the top of the exhaust valve guide. As a result, exhaust valve temperatures have been reduced as much as 200 deg. as indicated in tests.

TRANSVERSE SLOT in piston walls instead of the usual T slot previously machined is used in 1941 Buick. It is claimed that the transverse slot provides a stronger piston with less tendency to weaken and collapse. Pistons have a new skirt design which provides a larger area of contact on the piston surface for lubricating oil. New ribbing is provided under the top of the piston to increase strength. *(Continued on next page)*



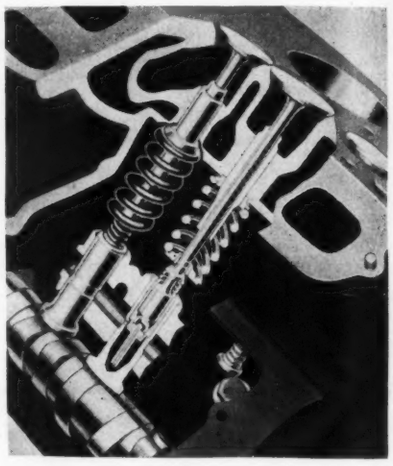
Oil-cleaning devices of the filtering type operate at maximum efficiency when new, but require filter replacement. Most filters operate on the by-pass system, oil from the sump passing through the oil pumps and then to bearings before passing through the oil filter. To pass oil through the cleaner before entering the bearings, and to maintain maximum efficiency indefinitely, at all engine operating temperatures and with any oil, Pontiac has developed a permanent-type oil cleaner. The cleaner consists of a settling chamber concentric with an oil inlet tube. Surrounding the settling chamber is an inverted cup; between the cup and settling chamber is a cylindrical screen. All lubricating oil going to the pump passes first through the screen to exclude any large particles of dirt. It then enters the inlet pipe and flows against the inverted baffle. This reverses oil flow direction, causing it to strike the flat baffle, and again reverse. At this point dirt particles are thrown out of the oil into the settling chamber where they precipitate on the bottom. After reversing direction as it strikes the baffle, the oil goes to the oil pump inlet tube connected to the die-cast cleaner cover, then to the pump and to the engine bearings. The cleaner is located on the suction side of the oil pump so that all oil passes through the cleaner before entering bearings. There is no filter element to dry or plug up, and the cleaner may be used efficiently for life of engine. Since cleaner is built in, there is no possibility of leaking tubes.

DOUBLE-ACTING SHOCK ABSORBERS on 1941 Buick models have arms which function also as upper control arms of the front suspension. The outer ends of the arms are secured to the knuckle supports with threaded steel pins and bushings. Front suspension is designed to permit easy caster adjustment. The boss forged on the outer side of the knuckle support locates and secures the king pin in the correct location. Chassis springs are held in position with a spring riveted to the lower control arm in the frame front cross member, and are insulated from the frame by fabric insulating shims.

TOP PISTON RING in the 1941 Buick is notched on the inner diameter to give it a slight rocking action. As the piston is forced downward, the lower edge of the ring is forced into the cylinder wall. The lower compression ring is taper-faced to present a sharp edge to the cylinder wall. This aids in reducing oil consumption during breaking-in period, and pomotes more efficient fuel consumption throughout the life of the engine. The lands in the piston and the piston rings are designed to decrease blow-by and to contribute to longer ring and piston life in addition to preventing burning of the top land.

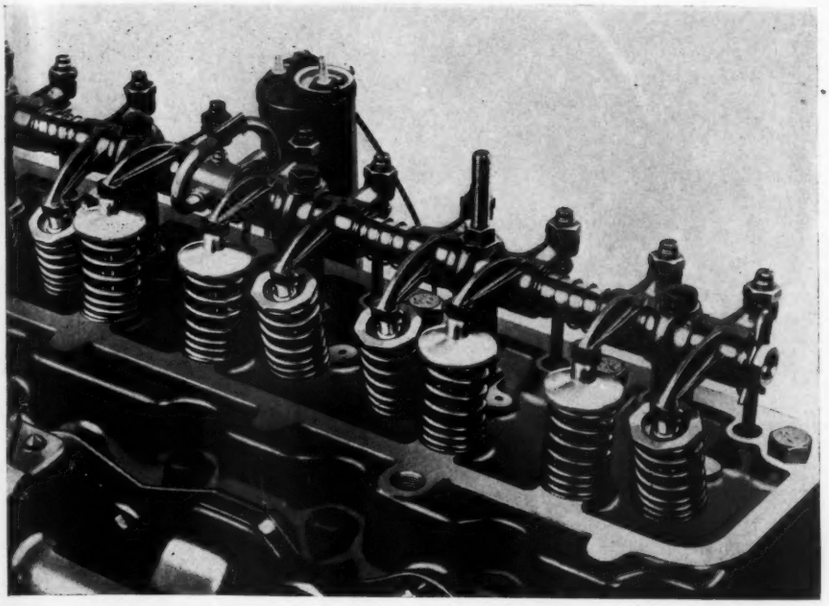
UNIFORM-WEIGHT CONNECTING RODS are used in 1941 Buicks. In all engines the rods have same weight and center of gravity. In addition to promoting smoother operation, this design makes the connecting rods in the same series interchangeable. The rods have been further strengthened by making the caps double-ribbed, increasing the rigidity at the cap bolts.

NICKEL-ALLOY ELECRO-PLATED PISTONS in Pontiac and Buick 1941 models have the piston pin slot at the side rather than at the bottom. Under advance spark conditions it was found that pistons which were assembled with the piston pin slot at the side rather than at the bottom would take 30 per cent more load before failure.



Improved hydraulic valve silencers designed for durability and smoother operation are used in the Cadillac 1941 cars. The valve lifter unit is now being manufactured from case-hardened steel, it was formerly cast iron. The new material, besides being more wear-resistant tends to prevent galling between the lifting unit and the body, thus permitting smoother operation and reducing wear. Tolerances between the lifting unit and the body are held closer to reduce oil leak-down and to assure more uniform valve lift. The lifter unit has been redesigned and the oil bleeder hole in the body has been relocated to permit a greater volume of oil to remain in the reservoir when the engine is at rest. As a result, the ratio of air to oil in the unit has been reduced and the effectiveness of the valve silencer on the engine has been improved.

COMPRESSION RATIOS in the Hudson "10" have been increased to 7.25 to 1, the 6-cylinder motor developing 92 hp. Spark advance has been changed to give an earlier spark at lower speeds.



Rocker arms of the 1941 Chevrolet, redesigned, are made of the newly-developed ArMaSteel, a special semi-cast steel having unusual wear-resisting properties that make possible the elimination of the usual bronze bushings. The arms are diamond bored to operate directly on the shaft. After boring they are Granodized to protect the bearing surface from scuffing or scouring when operating against the hardened rocker arm shafts.

SELF-LOCKING CONNECTING ROD BOLTS and nuts introduced in 1940 models are now used on Pontiac piston pin screw. Formerly the piston-pin screw was locked in place with a jam nut. The self-locking screw is effective; it reduces the number of parts.

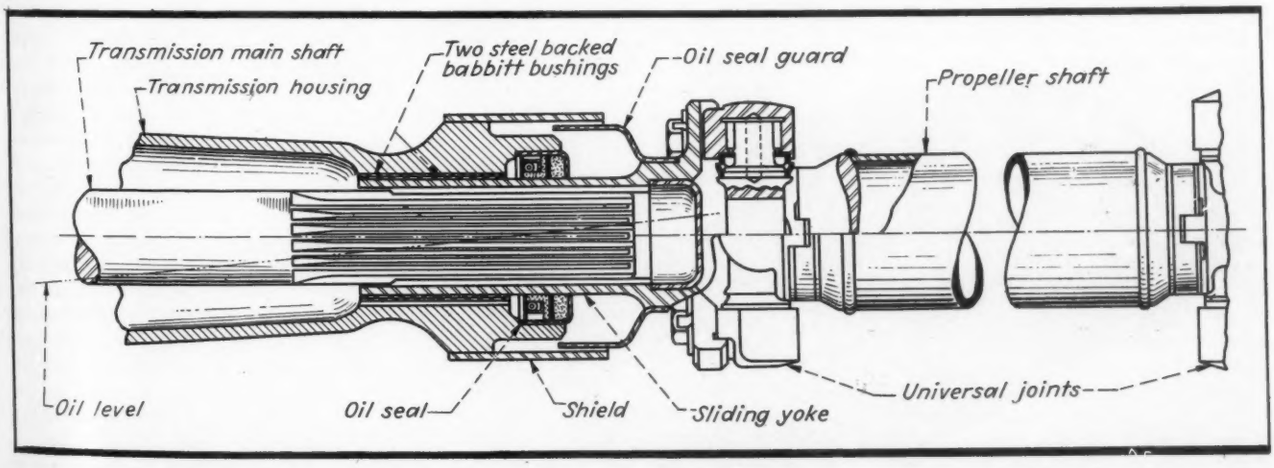
SILVER-PLATING of electrical contact points to prevent corrosion and to prolong trouble-free operation is continued in the 1941 Hudson models.

IMPROVED OIL FILTERING systems are on the 1941 Buick. Previously the filter outlet led to the valve overhead mechanism, necessitating a safety valve which functions when over 20 lb. per sq. in. pressure is required to force oil through filter. With sediment filter becomes ineffective. In the new system the filter outlet returns to the crankcase and there is no by-pass valve. This permits the oil to flow through filter at higher pressure, increasing filter life. If the filter becomes inoperative, oil remains in crankcase.

AN ALLOY CAST IRON CAMSHAFT replaces the former forged Cadillac camshaft. A special heat-resistant bearing material has been developed for the exhaust valve stem, this material being highly score-resistant at high temperatures. Exhaust valve head material is austenitic steel. Intake valves are chrome-nickel steel.

TRANSMISSIONS in the 1941 Hudsons are entirely new and are synchronizing geared. New transmission is 3 in. longer, all gears are helical. The overdrive has also been redesigned to eliminate a dozen parts. Overdrive also comes into operation at a lower speed than in previous models, at about 19 mi. per hr.

PONTIAC HAS INCREASED THE BORE of the 6-cylinder engine from $3\frac{7}{16}$ in. to $3\frac{9}{16}$ in., the stroke remaining 4 in. Total displacement is now 239 cu. in. per cylinder. This has increased brake horsepower to 90 over last year's figure of 87, while torque has been stepped up from 164 to 175 lb. ft. These increases in power and torque have been effected without increasing overall dimensions or weight of the engine. A slight change has been made in the contour of the piston head for the larger size so that the new piston weighs no more than the smaller diameter one used last year. Thus reciprocating forces will not be increased nor will connecting rod or crankshaft bearing loads be materially raised. Because of the larger displacement, it was necessary to increase the size of the intake manifold steel tube riser $\frac{1}{8}$ in. in diameter to handle fuel for full throttle operation. Dual carburetion is now standard equipment on all 8-cylinder Pontiacs.



Splined joint on propeller shaft is now removed from the propeller shaft assembly, and is located within the Cadillac transmission housing, assuring constant and complete lubrication. Splines are on the rear of the transmission main shaft. The ball bearing on the main shaft has been replaced by two steel-backed babbitt bushings pressed into the housing, which also holds the oil seal. These bushings are lubricated by the transmission oil.

Nash AMBASSADOR 600

Conventional sized, low-priced car with all-welded integral body and chassis

INVASION of the low-price field, dominated by the "big three": Ford, Chevrolet and Plymouth, was successfully attempted early last year by Studebaker with the new Champion, the engineering features of which were described in *P.E. June 1939, p. 226.* This year Nash enters the same field with an outstanding new design embodying features of size, performance and appointments customarily found only in the medium-price range. Previous Nash models covered the price range that represents 35 percent of the domestic market; with this new car, Nash's line for '41 will cover 92 percent of the domestic market. Based on this coverage, Nash expects to double 1940 sales this model year.

Faced with a highly competitive market, Nash engineers had to create a car that could be produced at minimum cost and yet would have distinctive plus features that competitors did not have. This was to be no "small" car: it was to be standard-size. It was to have "big car" performance and rideability. Most important of all, it was to be a real "economy" car in both first cost and operation. Appearance was to be similar to the remainder of the line, with no attempt to make the new model look like a stepchild or "loss leader."

After months of study, it was decided to develop an integral chassis frame body design with internal bridge-truss construction. Following aircraft design practice, roof, sides and floor were designed as formed, load-carrying members. In this way, it was possible to make the body of the new car practically the same size in cubic feet of interior space as the two larger Nash models. The entire body is welded into a single unit: no chassis frame is used. Almost 500 lb. of weight were saved by this type of construction. Additional production savings were made by providing interchangeability of doors and standard parts among the three lines.

Since body dead weight was considerably decreased by this design, it followed naturally that a small horsepower engine with all its attendant economies could be used. The new six cylinder engine develops 75 hp. at 3,600 r.p.m.; in details it follows Nash standards. Particular attention was given to economy. Uniform temperature control is obtained by casting inlet manifolds inside the head. Cylinders are completely water-jacketed. Compression ratio is 6.7 to one, bore $3\frac{1}{8}$ in., stroke $3\frac{3}{4}$ in. Carburetion is downdraft; pistons are steel-strut aluminum construction. Lubrication of all pistons, pins and cylinders is full-pressure type, and connecting rods are rifle-bored. Engine is fitted with a torsional vibration dampener. Entire crankshaft and flywheel assembly is dynamically balanced by electronic controlled balancer. Careful design has limited the weight of assembled engine and transmission, including usual accessories, to 460 lb.

Extensive road tests have shown the new Ambassador 600 capable of traveling well over 30 miles on a gallon of gasoline. Conservatively, the corporation is restricting advertising claims to 25 to 30 miles per gallon.

The body is sprung directly on coil springs on all four wheels and is the

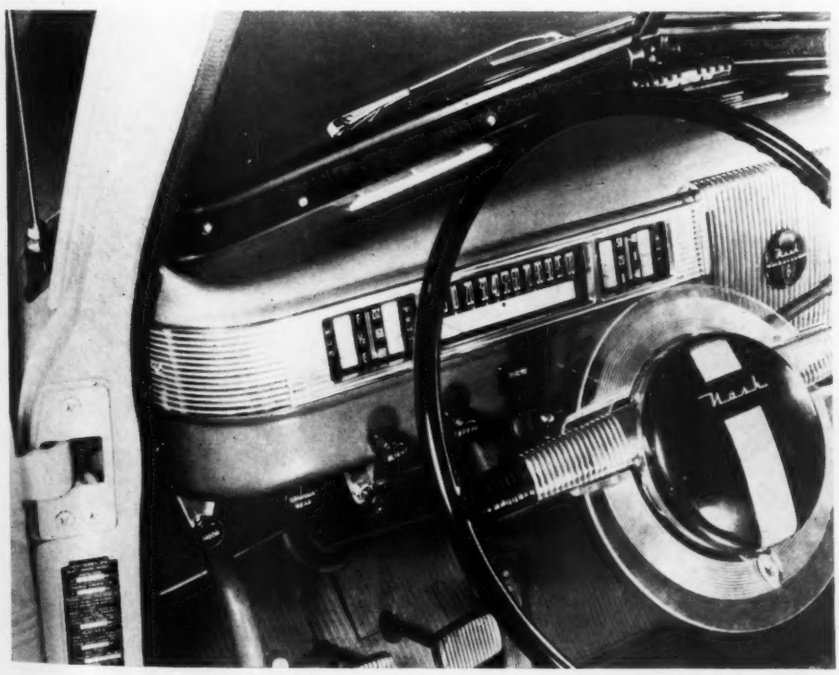
first low-priced American car to be so designed. The front end is carried on a Lancia type suspension; the rear is carried on coil springs nested into the integral frame. Direct-acting shock absorbers are mounted inside the coil springs at the rear of the car. Shock absorbers at the front end operate parallel to the kingpin.

The front end suspension may be described as a floating kingpin construction. The kingpins—20 in. long—are set vertically and supported at the top by a horizontal, steel-tube cross rod. Kingpin is supported at the bottom by a heavy, formed, bridge-type truss member welded to the steel body. Front wheel spindle is mounted on a mirror-smooth collar which moves up and down the lower section of each kingpin on six caged roller bearings. The upper end of this collar supports a resilient coil spring which in turn carries the body weight on a tapered roller bearing. The entire assembly is pre-lubricated and fitted with felt grease and dirt seals; it is completely housed in a corrugated rubber boot.

With the low unsprung weight in this design and direct soft coil spring action, the new car offers a remarkably smooth ride. An additional advantage with floating kingpin spindle mounting is that it permits a two-way ball bearing steering system with consequent reduction in steering effort. The car will swing in a 33-ft. diameter circle.

Bodies, which are all torpedo models with the choice of slipstream or trunk back, are completely insulated and sound-proofed with "Sand-Mortex" applied to body panels. Doors are weather-sealed. Both body and fenders are Bonderized, and are finished in slow-baked high-gloss Permalux enamel. Glass area on the new car is increased over the largest 1940 Nash, the windshield having 88 more sq. in. of area and the curved rear window having 40 more sq. in. of area. Steering wheel gearshift is furnished.

(Continued on next page)



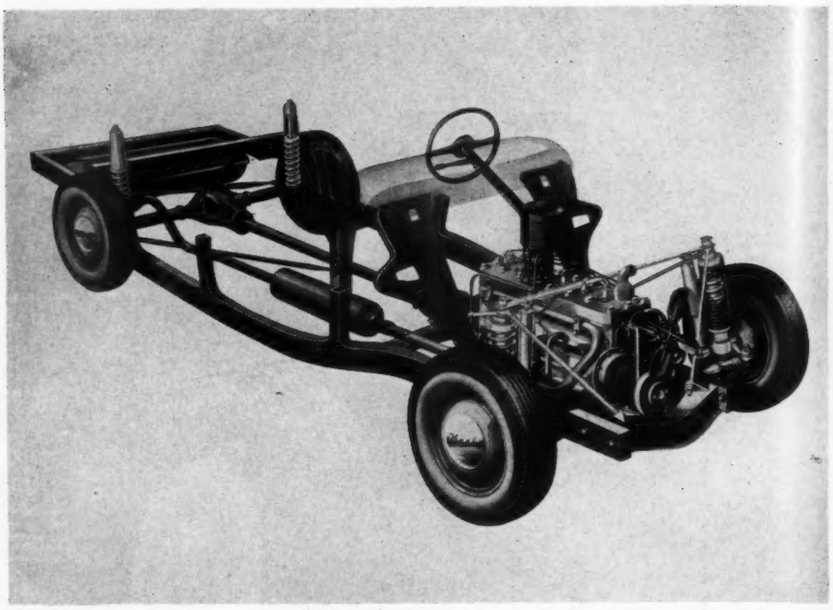
Steering wheel and instrument panel of new Nash. Wheel, centerpiece and control knobs are red Tenite. Horn ring is molded of clear Crystalite.

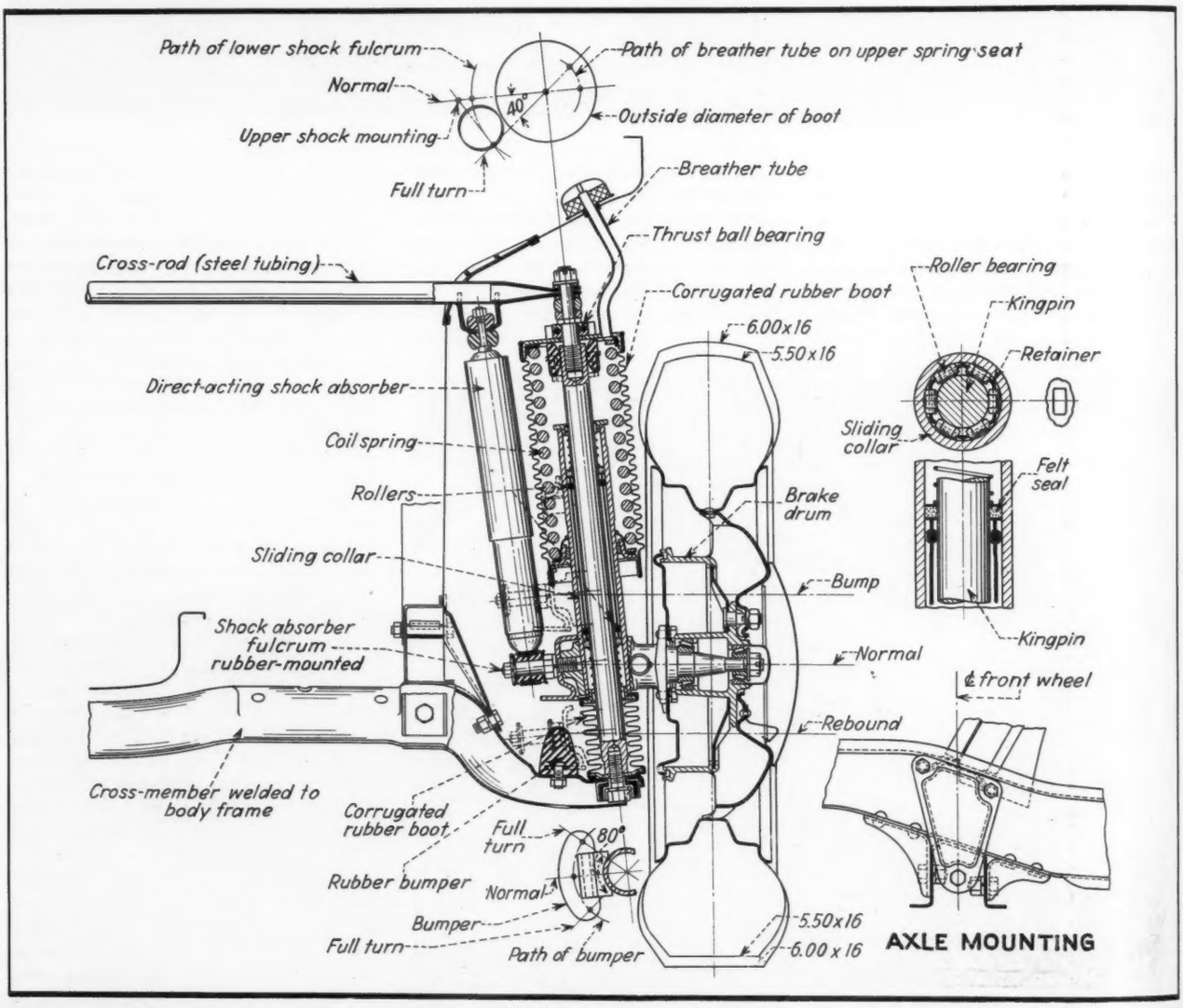
NASH AMBASSADOR 600

Constructional details

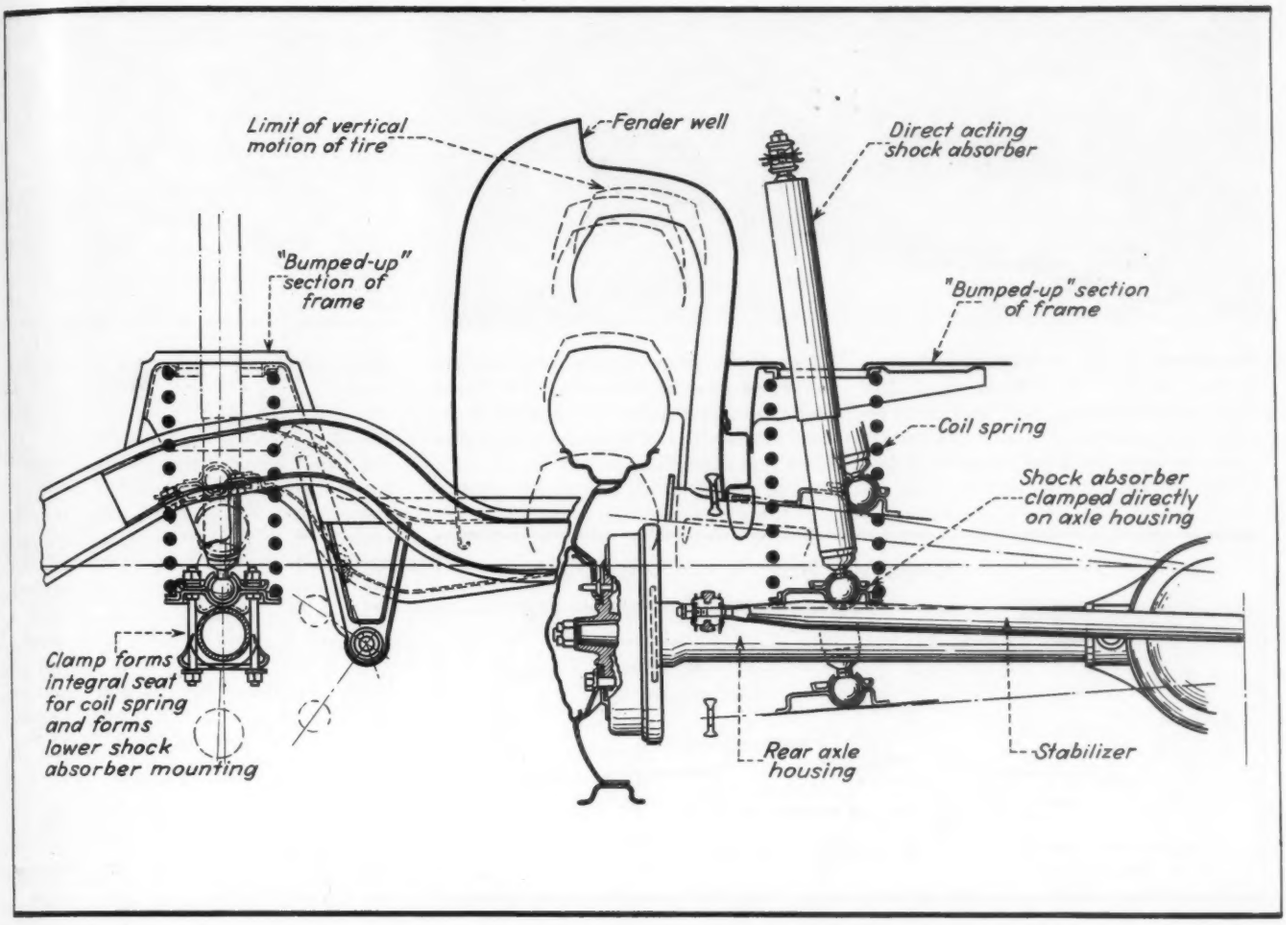
SHOWN AT THE RIGHT is a phantom of the new 600 method of springing. The front suspension was developed from the original Lancia racing car designs and provides knee-action operation and advantages with a minimum number of parts, since upper and lower control arms and associated mountings are not required. Camber adjustment is much simpler than with conventional knee-action suspensions. The rear coil springs are housed in a formed box section; lower ends are bolted to axle.

An inclined member carries load from the front supports directly to the body frame and helps distribute stresses to the body shell. This bridge type truss member is also being used on the regular Nash series.

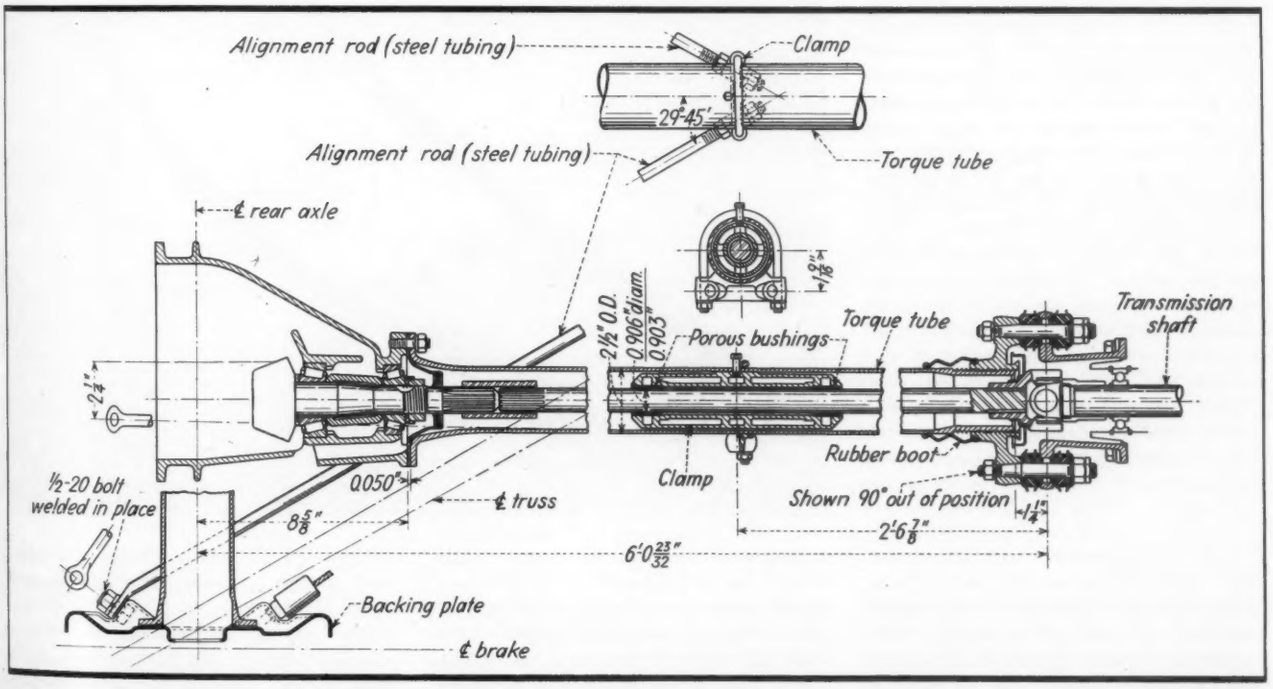




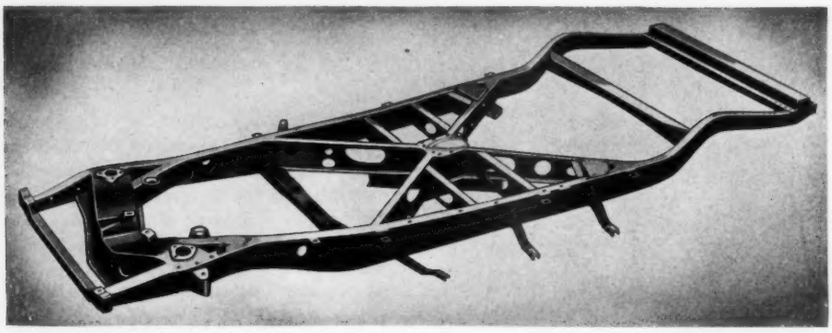
Floating kingpin design embodies a sliding collar directly connected to the wheel spindle and operating against a coil spring. Special-shaped roller bearings give roller action in two planes. Body mounts on ball thrust bearing.


Limit of vertical motion of tire
Fender well
Direct acting shock absorber
"Bumped-up" section of frame
"Bumped-up" section of frame
Coil spring
Shock absorber clamped directly on axle housing
Clamp forms integral seat for coil spring and forms lower shock absorber mounting
Rear axle housing
Stabilizer

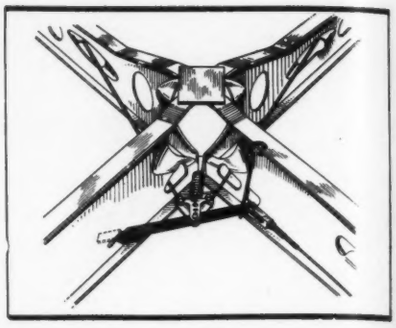
Direct-acting shock absorbers are mounted inside the rear coil springs. Lower mounting of spring is directly bolted to real axle housing. Rubber bumpers absorb shock on extreme compression or elongation of spring.



Torque-tube clamp constricts tube and holds center bearings in place; it is fitted with porous bronze bushings which position axle. Rubber mounted alignment bars extending from rear axle to torque tube provide lateral cushioning.

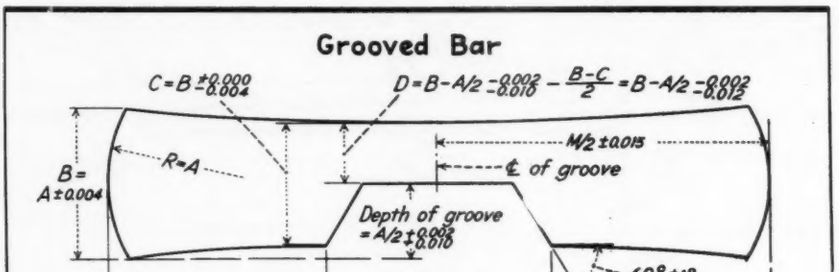


Armored X-type frame for convertible models in the 1941 Hudson. An extra sub-*X* member, with a heavy reinforcing plate and four additional struts connects the *X* members with the sides of the frame, giving a total depth of 9⅞ in. of reinforced steel. The frame for closed models is rigid enough without the two parallel lateral cross-members and bi-sections of the *X*.

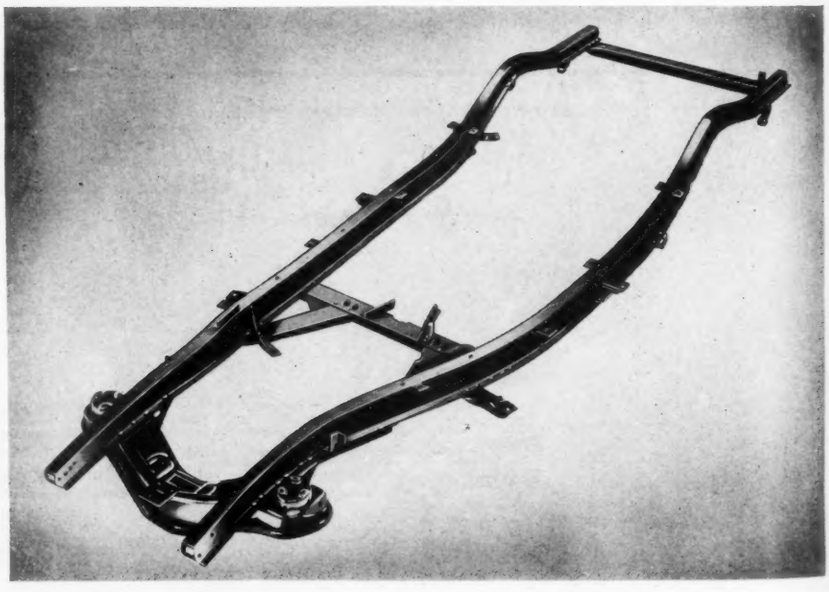


Improved emergency brake control on 1941 Pontiac. Horizontal link was supported in 1940 models in horizontal slots in the *X*-member webs. During intermittent freezing and thawing, ice formation between link and slots made release of brake difficult. To overcome this tendency to freeze up, the new design retains the pivoted end, but the opposite end does not project through the *X*-member web. Instead a wire link is used. As the link no longer rests in the *X*-member slot, ice has no chance to form around it, and so far as the pivoted end is concerned, enough force is available to break loose any ice which may form.

To INCREASE RIGIDITY, the lower edge of the Pontiac instrument panel is rolled up. This latter operation also conceals the raw metal edge.

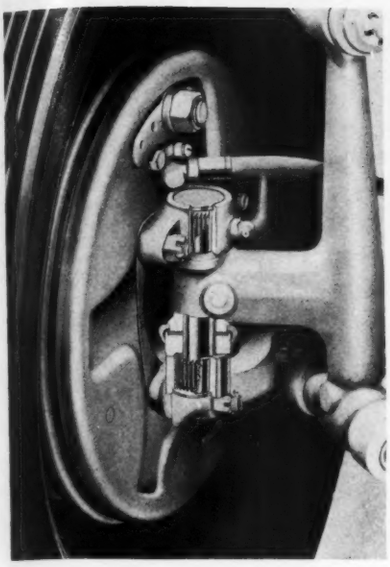


Grooved spring leaves of Amola steel are now standard equipment on Chrysler Corporation cars. Leaves are grooved on compression side. Grooved section has lower maximum tensile stresses for same loading because neutral axis is moved nearer tensile edge. To provide interchangeability of present and former springs the same spring rate is attained by using a thicker section grooved leaf that has the same total moment of inertia. The increased depth for the same number of leaves is taken up in axle clips. A saving of 8 to 9 percent in steel weight and reduction of 2 to 3 percent in tensile stress is obtained with a grooved leaf spring as compared with flat leaf spring of the same rate. It has been found that shot blasting the surface that is under tensile stress increases the fatigue strength; this shot-blasting operation is performed on longest four or five leaves only.

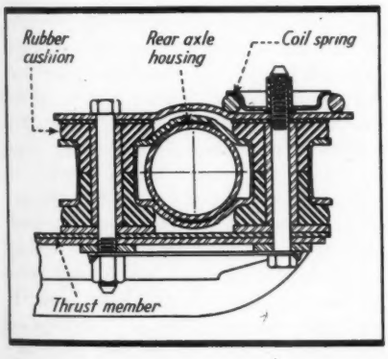


VACUUM WINDSHIELD WIPER action has been improved on the 1941 Ford. Vacuum connection, formerly on the intake manifold, took vacuum principally from cylinders number 7 and 8. The new connection is located on the carburetor riser and obtains its supply from all eight cylinders.

Chevrolet frame has been lengthened 3 in. and lowered ⅝ in. Widened frame prevents excessive overhang of the wider body, keeps the length of the body brackets down to a minimum. Greater rigidity is obtained in mounting of body to frame by the use of additional two-bolt attachments, and extra body bolts in new locations. With the new design, two-bolt attachments are used at the dash, and at each of the three body pillars on each side. Extra one-bolt mounts are provided on each side midway between the first two pillars. This construction is effective in increasing the floor stiffness.

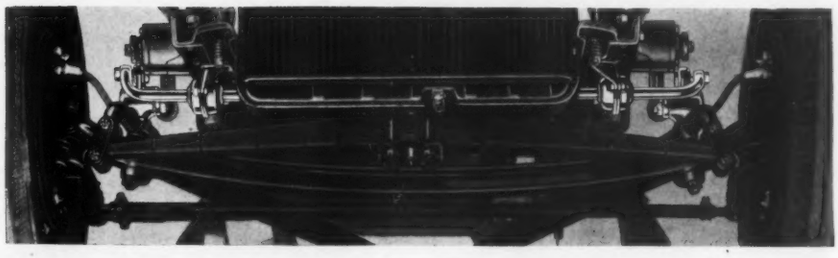


Vertical front wheel support in the Packard is a steel forging carrying the front wheel spindle and yoked to the upper and lower support assemblies. The lower-support lever assembly is hinged to the vertical wheel support by a horizontal king pin mounted on two roller bearings. Needle bearings have been installed at the top and bottom of the steering knuckle king pin. The ball thrust bearing used in 1940 models has been replaced by an antimony-lead thrust bearing.

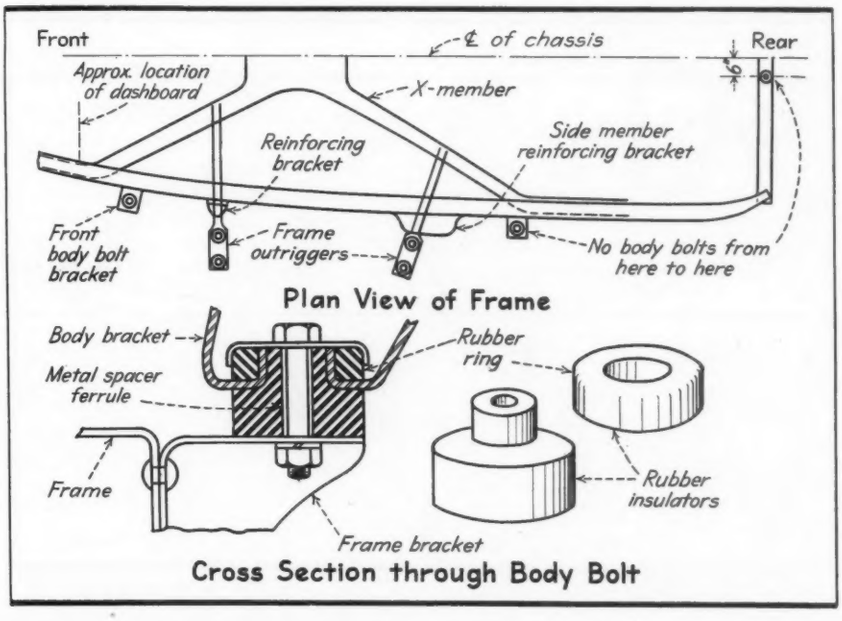


Rear support for thrust member in Oldsmobile has been redesigned. New support consists of a formed bracket welded to rear axle housing. Thrust member is supported on rubber cushions in this bracket. Mounting is bolted together for easy assembly.

BECAUSE THE TRUNK FLOOR IS FLAT all the way to the door opening it was possible to weld a reinforcing channel across the rear edge of the floor and reduce drumming noises in the Pontiac. A heavy gage steel gutter is welded entirely around the trunk door opening to increase rigidity.

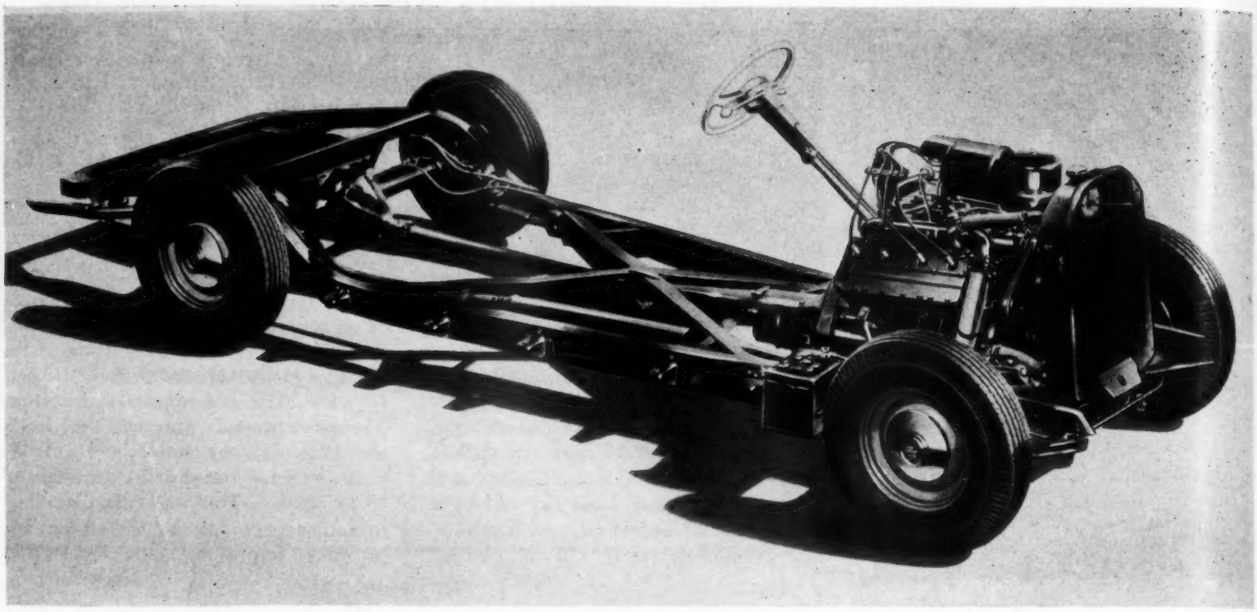


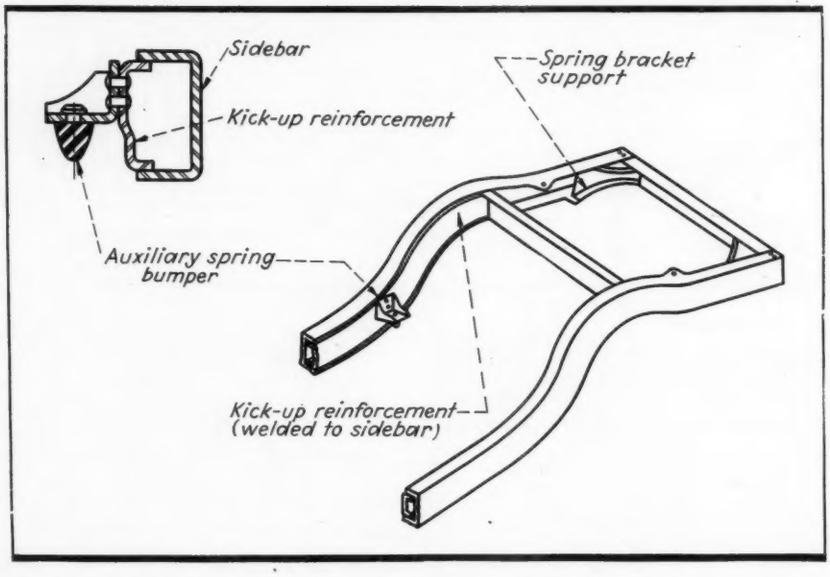
Front spring frequency on Ford has been lowered 16 percent, camber has been reduced one inch and the deflection rate has been lowered. 1940 front spring frequency was approximately 76 oscillations per minute; the frequency of the 1941 front spring has been lowered to 63.7 oscillations per minute. The former front spring had a 185-lb. per in. deflection rate. The new rate is approximately 175 lb. per in. Lower spring frequency is obtained by use of new reversed type spring shackles. On the 1940 car the lower ends of the shackles sloped inward toward the center of the car and approached vertical position with full passenger load. On the 1941 car the lower ends of the shackles slope outward at an angle of 25 to 30 deg. This puts the main leaf in compression instead of tension and provides improved damping action.



Improved method of mounting the Ford 1941 bodies to prevent road strains being transferred to the body structure. The fourteen rubber insulators for the body bolts, used for all body types except the convertible club coupe, are twice the size previously used. There are no mountings near the rear corners of the body. Two rear bolts are on the frame cross member; they are spaced 6 inches on each side of the longitudinal center line of the car. Frame weave at the rear cannot be transferred to the body structure. Rubber insulation for the body bolts consists of two pieces. A metal sleeve or bushing on the bolt prevents the insulators from being compressed beyond a predetermined amount when the body bolts are tightened. Thus each insulator has the same degree of resiliency; this insures uniform action from all of the mountings. In the convertible club coupe, metal spacers replace the rubber insulators to provide increased rigidity for this type body.

HOTCHKISS DRIVE REAR SUSPENSION with permanently lubricated rear springs are continued in the Cadillac line. Rubber bushings on the rear spring shackles insulate the frame and springs at all points of suspension. All series are equipped with an auxiliary rubber spring bumper, mounted on the sidebar kick-up reinforcement, to cushion spring action over rough roads. Wax-impregnated liners are used. *(Continued on next page)*





Greater rigidity in 1941 Cadillac frames (above and left) is obtained by new X-member and sidebar construction. Lighter gage sidebar stock is used, but depth has been increased and reinforcements added. A channel section reinforcement securely welded to the frame sidebar extends along the side-bar from the rear of the X-member to the fuel tank cross member. This forms a rigid support to the kick-up over the rear axle. Support, added at the junction of the rear cross member with the sidebar, is a heavy Z-section reinforcement at the rear spring bracket. Greater stability is obtained through increased frame width at the rear and increased rear treads which place points of spring suspension further apart. Front end rigidity has been increased by joining the X-member to the sidebar at a point farther to the rear of frame. A long channel extension forms a box section with a sidebar from this junction to the front cross member. These added improvements and the deeper sidebar construction have added about 40 percent more torsional stiffness to the frame used for sedan and coupe models.

TO INSURE GREATER STABILITY in the steering column and wheel, the thickness of the column jacket has been increased from 0.049 to 0.085 in. in the 1941 Pontiac. Doubling the metal gage increased the stiffness of the tube approximately four times, thus reducing vibrational quiver.

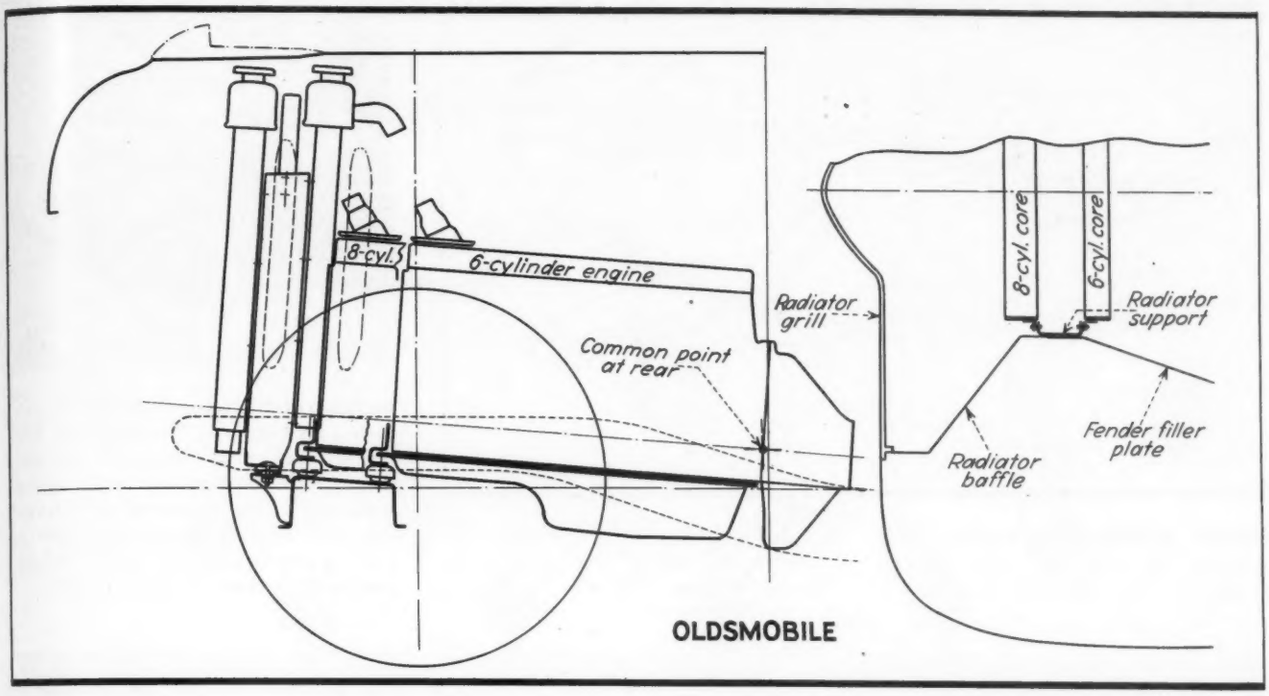
FRONT SUSPENSION of the Series 90 Buick is simplified by having the kingpin support boss forged integral with knuckle support. This replaces caster adjuster assembly previously used. Direct cross steering tie-rods eliminate intermediate steering arm and bracket assembly used in 1940. The new steering tie-rod assembly connects directly from steering gear pitman arm to the steering arms. Assembly is simplified.

IN THE 1941 BUICKS the series 40 is built on an improved 1940 series 50 frame and the series 60 is built on an improved 1940 series 70 frame.

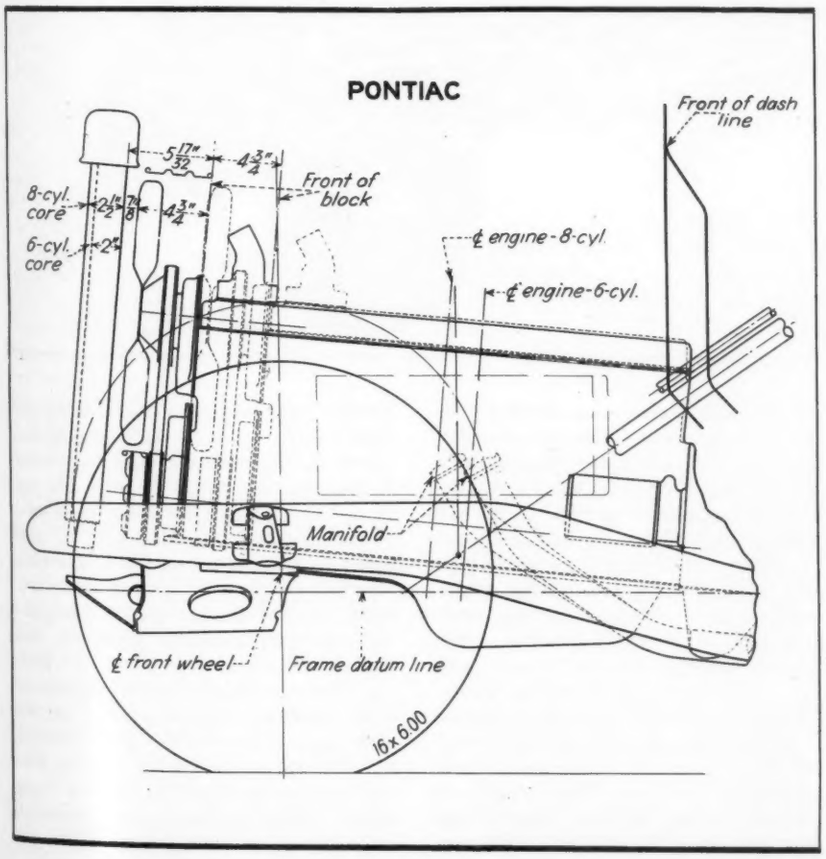
STEERING GEAR WORM is burnished in the 1941 Pontiac. With the usual machined surface, tooth pressure ironed out the ridges left in the machining operation, increasing clearance between roller and worm and adding backlash in the steering wheel mechanism.

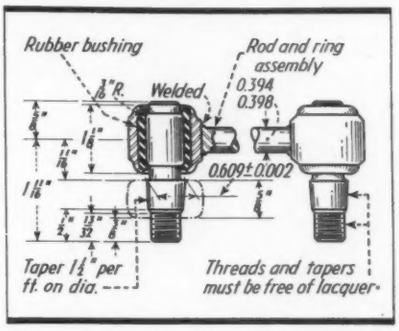
ALL-STEEL WELDED CONSTRUCTION is used for seat frames instead of the usual wood bottom frame in the 1941 Chevrolet. The new construction, using steel tubing and a deeply ribbed stamped steel section, gives a maximum of strength for a minimum of weight.

AN ALL-STEEL DRUM 1/8 in. thick with a braking surface of centrifugally-cast nickel alloy iron replaces the cast drum formerly used on Pontiac. This steel shell protects the cast iron braking service from breakage. After the shell is pressed to shape it is placed in a revolving mold and a stream of close-grained nickel-alloy molten iron forced against the inner cylindrical surface, to produce an iron liner of uniform hardness which machines smooth and provides an excellent braking surface.



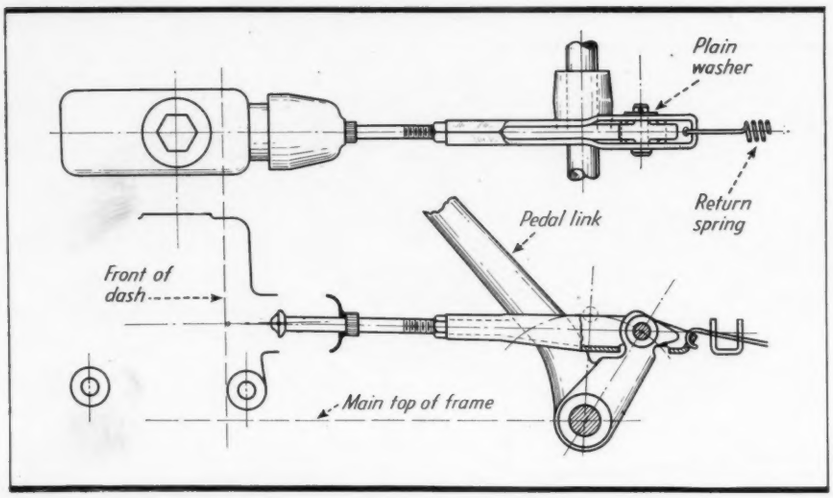
Same frame for sixes and eights. Two methods used in 1941 cars to adapt same size frame for both six and eight cylinder engines. Oldsmobiles have two positions for mounting radiator core. See above. Rear engine mounting is in same location for both sizes of engine. Difference in length is taken up by method of mounting radiator core. With core in forward position, clearance is allowed for eight cylinder engine; when six cylinder engine is used, radiator core is mounted in reversed position. In the Pontiac design, below, the same frame also serves two engines. Again the rear engine mounting is in the same location for both engine sizes. Radiator core is not movable, but a fan shroud is used between engine fan and radiator. Cooling efficiency was so improved that a shroud was also used on the eight.



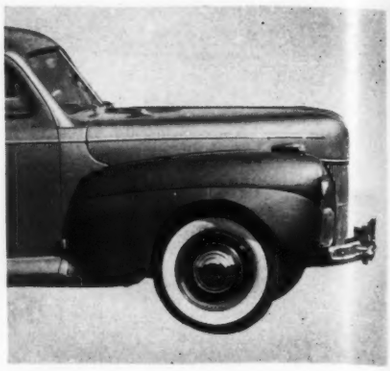


Rear shock absorber link construction in 1941 Buick reduces flutter of rear wheels on rough roads. Quick reaction to road surface conditions was not obtained with previous type links because the large rubber grommets allowed a certain amount of play in the link. In the new link, the stud at each end is set at a right angle to the link rod and is insulated with rubber.

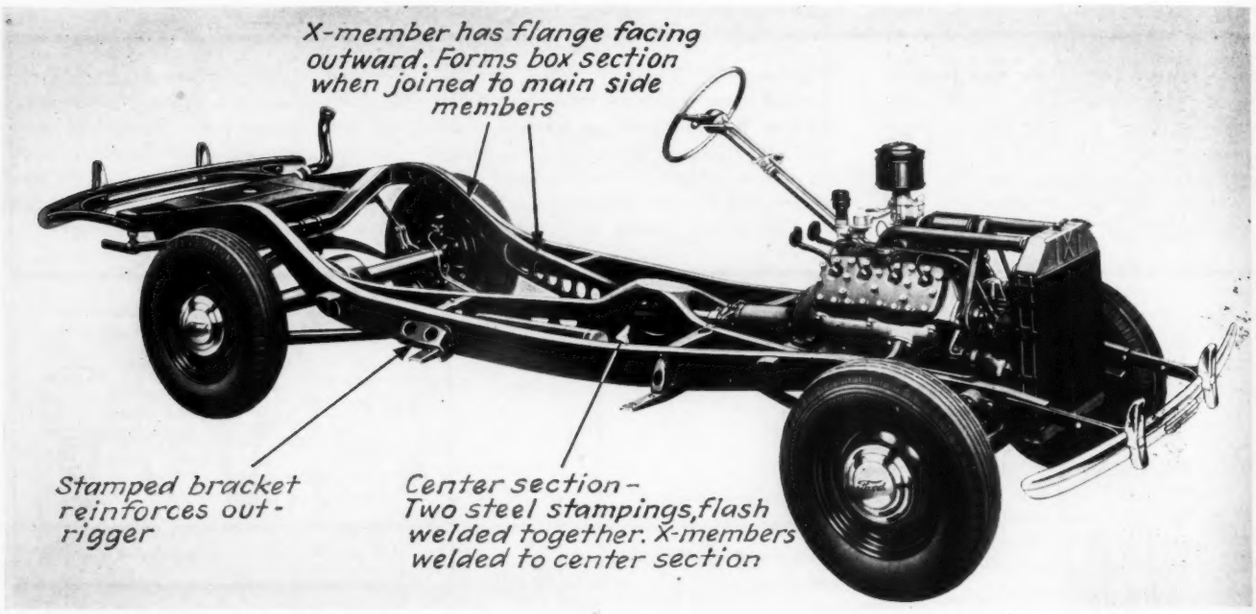
ALL CADILLAC, AND BUICK series are now equipped with the Saginaw recirculating ball type of stearing gear described in last November's article on page 465. The balls provide practically frictionless operation of the steering gear mechanism and reduce steering effort. (*Continued on next page*)

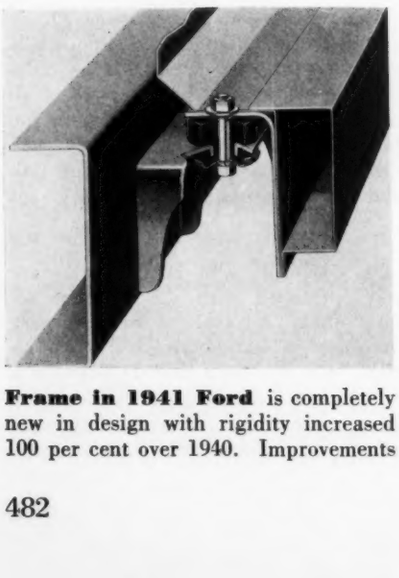


Brake cylinder actuating link is connected to the brake pedal through a pin, but link operates in slot as a safety feature in the new Oldsmobile. Pressure will still be transferred to hydraulic cylinder, even with pin failure.

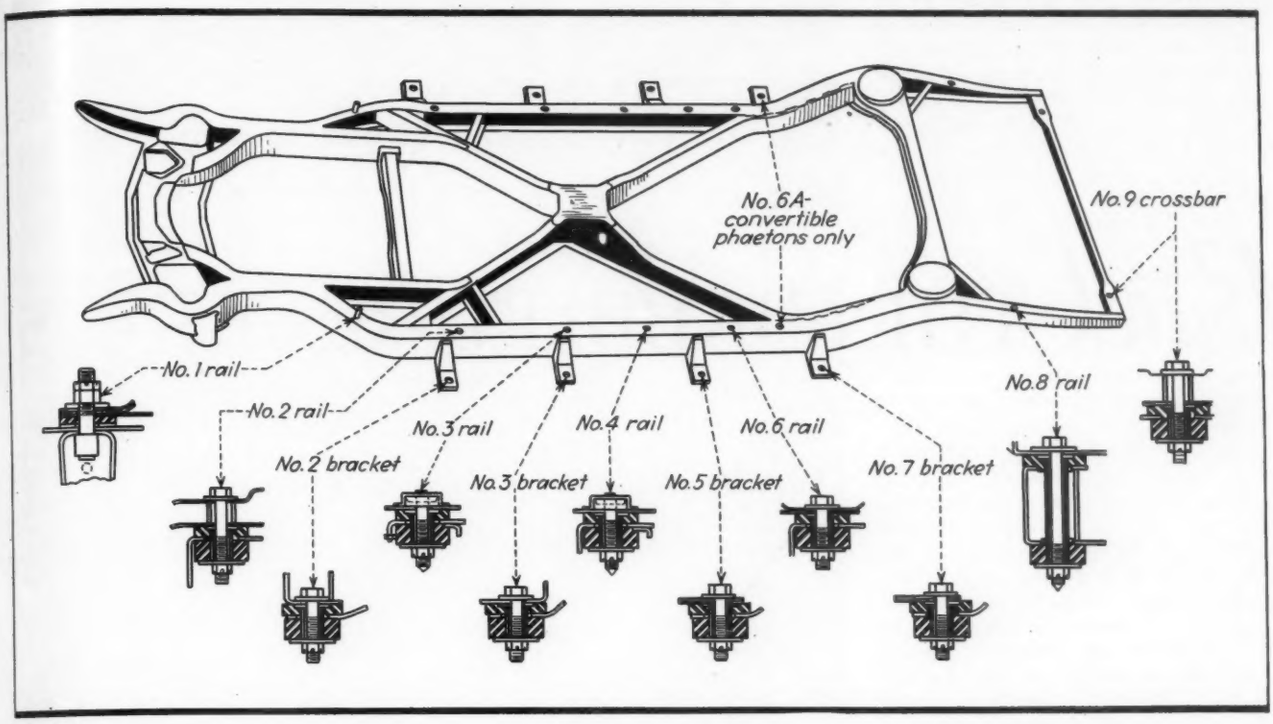


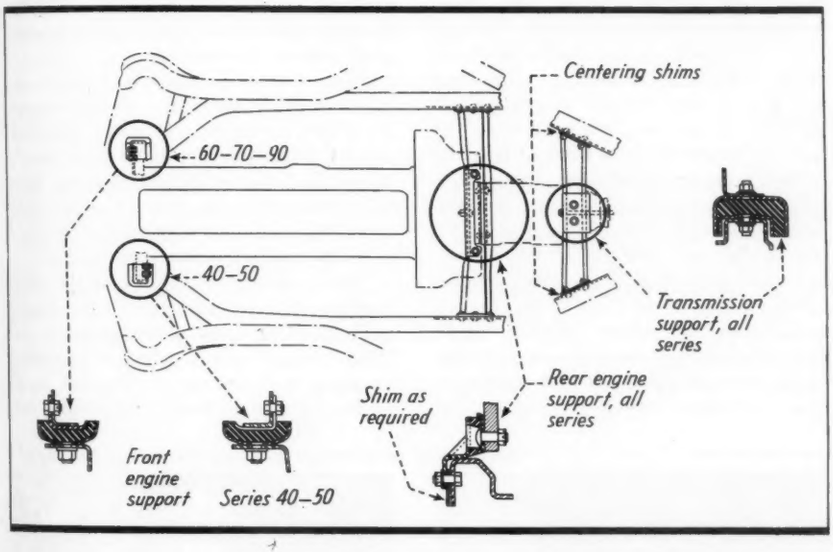
Multi-unit fenders are now furnished on Ford and Mercury cars. The fender, formed in three separate sections, is assembled with bolts and spring washers. Besides reducing metal-forming costs, high on deep-drawn fenders, sections can be replaced easily. Deluxe models have stainless trim on joint.


X-member has flange facing outward. Forms box section when joined to main side members
Stamped bracket reinforces outrigger
Center section - Two steel stampings, flash welded together. X-members welded to center section



Frame in 1941 Ford is completely new in design with rigidity increased 100 per cent over 1940. Improvements include a new *X*-type cross member with 9-in. deep box-type center section. The *X*-member is braced to the side members at 8 points. Torque tube now passes through the center section of the *X*-member. Since opposite front and rear *X*-member channels are in a straight line, torsional resistance is increased. Center section consists of two steel stampings, flash-welded together; the channels of the *X*-member are each flash-welded to this center section. These *X*-member channels have their flanges facing outward toward the side members, when joined to side members they form wide box sections. This design was first used on the Mercury frames last year. The channels on the new frame extend past the front and rear cross members to which chassis springs are attached. The unit structure is braced to the side members of the frame by eight additional transverse members. These members are deepened at the point of attachment to the frame side members. Where the two outriggers extend from the frame on each side, special stamped brackets reinforce the frame side members to minimize twisting. The rear frame cross member in cross section is an inverted channel with flanged legs with a plate welded to its flanges to form a box-section. The rear spring is U-bolted against this plate. Rigidity has been increased with only slight increase in weight.





New type body mountings have been developed for Buick cars to limit shim compression. Body mountings are of positive preloaded type; they have a spacer which limits the amount of compression that can be put on the rubber shims. This limit was established after a series of tests. This type of bolt is used on all except the front body bolt. Softer lower shims are now used on all bolts. Certain of the frame to body shims are round because it was found during tests that reduced shim area improved quietness. There is no metal-to-metal contact between frame and body, but rigidity is obtained primarily through the manner in which bolts are distributed along the frame. The plane of separation between frame and body serves as a wall of insulation against road noises and vibration.

NEW REAR SPRING on 1941 Ford is 1½ in. longer than formerly. The 1940 spring was clamped to the frame cross member for a distance of 6½ in. By redesign of the clamp, this distance has been reduced to 4 in. resulting in an additional 2½ in. of effective length. This makes a 4 in. increase in effective spring length. Reversed shackles help reduce rear spring frequency to 57.7 oscillations per minute. There is a difference of only 6 oscillations per minute between front and rear springs. This close balance between front and rear spring rates minimizes excessive pitching of the car on rough roads. Lighter shock-absorber settings can be used; this also improves ride.

TRUNK FLOOR BOARD in Pontiac extends across the body, increasing the rigidity of that section of the floor. This change requires all servicing of the clutch and transmission to be done from beneath the floor level. Rear doors are hinged on the center pillars which are made of larger sections. Box-typed concealed hinges with graphite impregnated bronze bushings are used on all bodies.

CAMBER ADJUSTMENT in 1941 Cadillac is made easier through replacement of threaded pin in upper end of steering knuckle support with a plain eccentric pin. Camber adjustment range has been doubled with this new design.

FRONT SEAT FRAMES in the Pontiacs are made entirely of steel, reinforced with a steel tube which follows the top edge of the steel framework.

WAX-IMPREGNATED LINERS are used between all spring leaves, except the lower three, in the Chrysler Imperial rear springs. Besides serving as a dry lubricating method requiring no owner attention, these liners permit control of leaf friction. Spring covers are eliminated. The liners between top leaves have ends protected by fabric sleeves which are steel-stapled to the liner. In addition the end of the liner is further protected by a crimped-on metal flange.

Buick THIN-BABBITT BEARINGS

Babbitt Bonded to Sintered Powder Metal Matrix, Steel-Backed

NEW THIN-BABBITT BEARINGS are one of the most significant developments in the 1941 cars. Bearings have been given particular attention by designers, because with the new high-octane fuels and higher piston loadings, increased horsepower and torque can be developed from a given size of engine. As a further step in this direction combustion chamber and piston head contours have been redesigned generally to obtain higher compression ratios.

Basic characteristics of all the new bearings is the thin babbitt coating, usually 0.005 in. or less thick. Chrysler, DeSoto, Dodge, Plymouth and Pontiac use such a bearing, steel-backed, in place of former types which had from 0.015 to 0.025 in. babbitt on steel backing. Dodge last year used a tri-metal bearing: thin babbitt, bronze and steel. Buick's new bearing has high-lead babbitt 0.001 to 0.002 in. thick, bonded to a sintered powder matrix which is permanently bonded to a steel back.

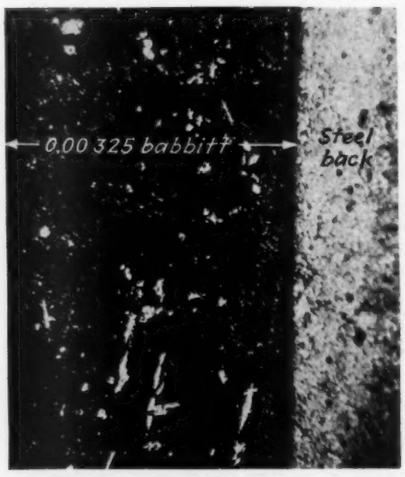
The Buick precision-type, steel-back bearing was developed to meet the basic requirements of a porous matrix or foundation firmly bonded to the steel backing, carrying a corrosion-resistant bearing material which would impregnate the "spongy" matrix, thus developing a bond both metallurgical and mechanical in nature.

Experimental exidence and service history indicated that high-lead babbitt alloy possessed excellent resistance to wear and corrosion. With the alloy relatively soft, wearing down of crankshaft journals and pins would be prevented, thus retaining the original bearing fits and, consequently, maintaining fine balance and smooth running. Although high-lead babbitt material has been known and used for many years, its most useful properties have been offset by a tendency to fatigue failure. Service history showed unmistakably that fatigue failure, evidenced by cracking and actual breaking out of portions of the bearing surface, was related to the use of a relatively thick layer of babbitt.

The superiority of these new thin-babbitt bearings lies in their greater fatigue strength. Studies revealed that practically all fatigue cracks were about the same depth. Incipient fatigue failure begins in the form of fine cracks or fissures in the surface, which progress radially inward. Eventually, at a certain depth these radial cracks are joined by short, so-called circumferential cracks parallel to the bearing surface and deep in the structure of the metal. This network of cracks results in the loosening of large flakes of metal which tear or melt out of the bearing.

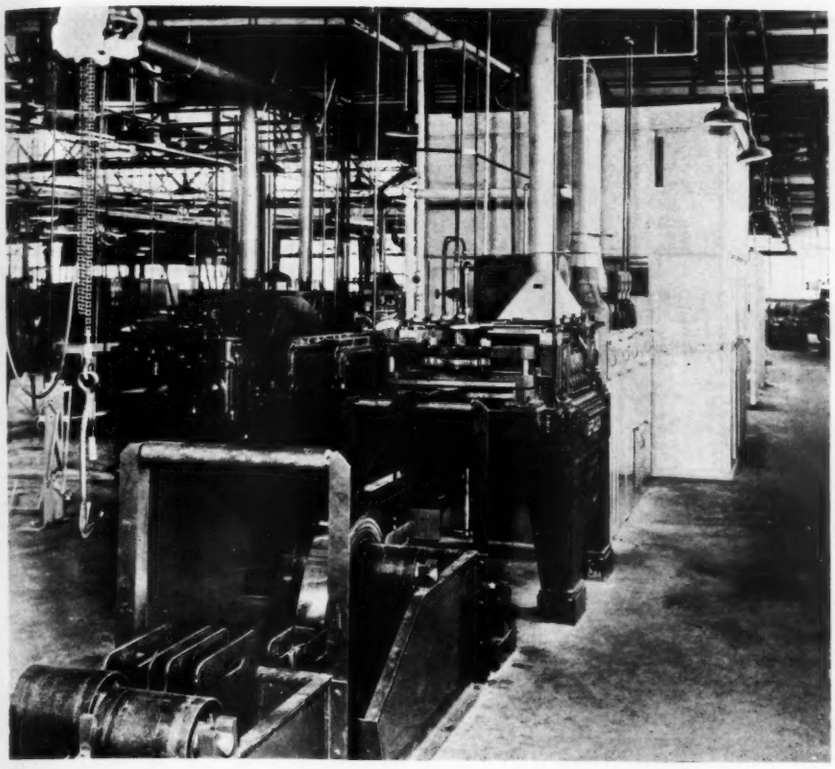
With extremely thin layers of bearing material, incipient fatigue cracks no longer developed. Moreover, if radial cracks did develop under certain conditions, the layer of babbitt was too thin to allow sufficient plastic flow that might cause the development of the connecting circumferential cracks.

The matrix for the Buick bearing consists of pure powders of copper and nickel, sintered at a temperature of 2,000 deg. F. to produce a durable bonding of the matrix to the steel back in the form of an actual chemical bond



Thin-babbitt bearing used in Chrysler line. Babbitt is bonded directly to steel back. Pontiac also uses same type.

Buick bearing—60 diameters—showing the thin babbitt anchored in the powder metal matrix bonded to steel.



Steel strip received in reels is made ready for the matrix powder by passing through the roller leveler machine in the foreground. The large booth in the background is the powder room where the matrix powders are mixed intimately and fed onto the surface of the moving strip.

which becomes a part of the steel strip. The strong sponge-like structure containing interlaced pores or voids and hook-like ends is impregnated with high-lead babbitt of 0.001 to 0.002 in. thickness. Since the inertia loads of a modern high compression, high performance engine may exceed 2,000 pounds per square inch of projected bearing area, the steel strip with its bonded matrix is pre-loaded during manufacture.

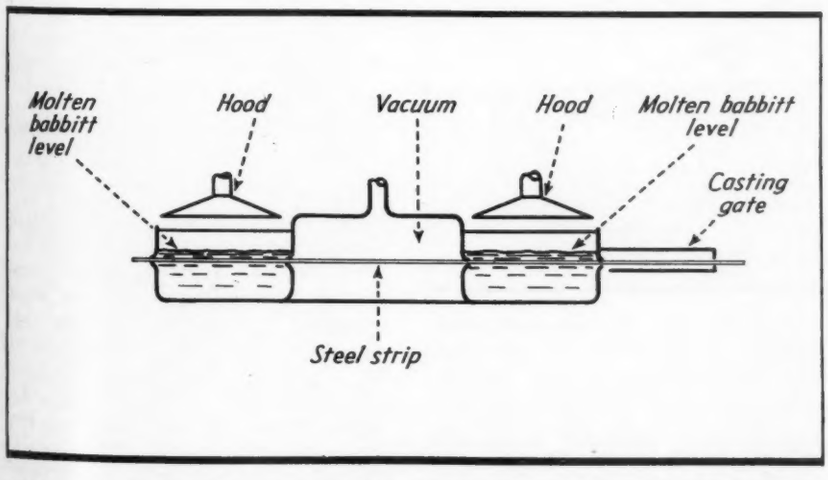
Bearing Production Methods

To produce this precision bearing, a special manufacturing division, embodying three separate lines, was set up by Durex. The first line is for the application and sintering of the matrix, the second for the formation of the babbitt coating and the preparation of blanks ready for the final forming and machining operations, and the third is the conventional set-up for finishing formed half-shells.

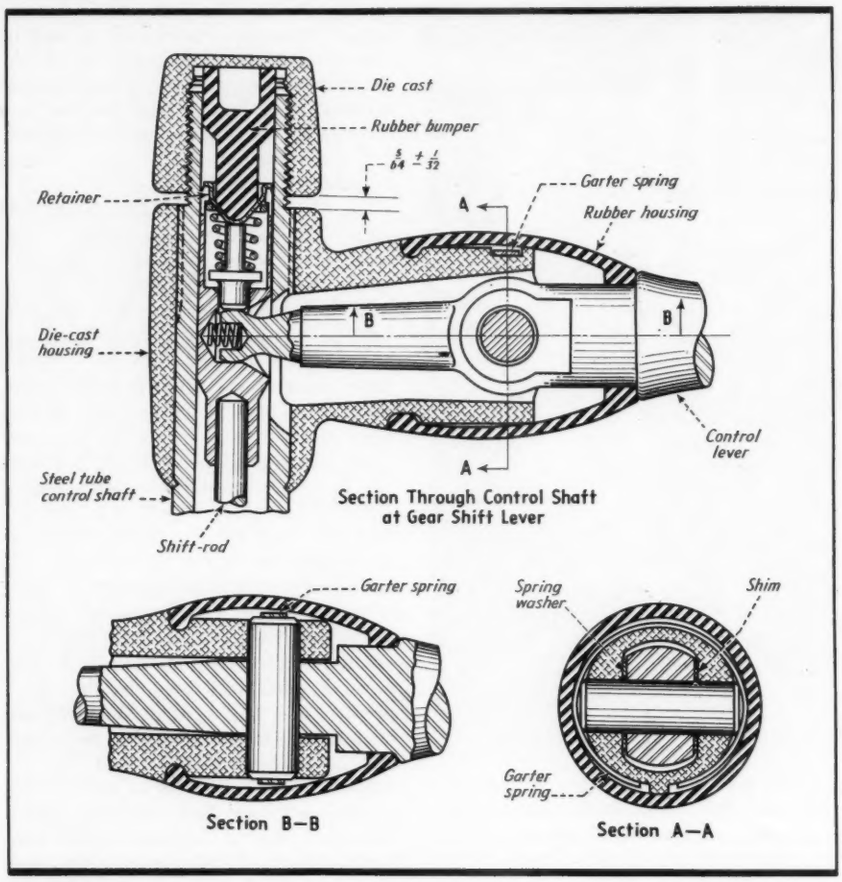
In the first line, *S.A.E.* 1010 cold rolled strip with thickness held to close tolerances, is fed through a roller leveller, then through an alkaline wash and drying machine to clean the surface of dirt and oil. From the washer, the strip is carried into the powder room where the intimately mixed powders of 60 percent copper and 40 percent nickel are fed onto the continuously moving strip. As the strip leaves the booth, the steel back passes over a brush which removes traces of powder and dirt from the underside. The powder is sintered in a special electric furnace in a prepared atmosphere at 2,000 deg. F., resulting in a porous matrix of controlled metallurgical character, bonded securely to the steel back. The powder particles, fused to the steel, produce a strong, sponge-like structure containing interlaced pores and voids with hook-like ends.

Reels are joined, end to end, by resistance welding. Next the strip passes through a set of chromium-plated embossing rolls which compress the matrix under loads as much as 10,000 lb. per sq.in., consolidating the powder-metal sinter, giving it uniform thickness and achieving a permanently pre-loaded structure.

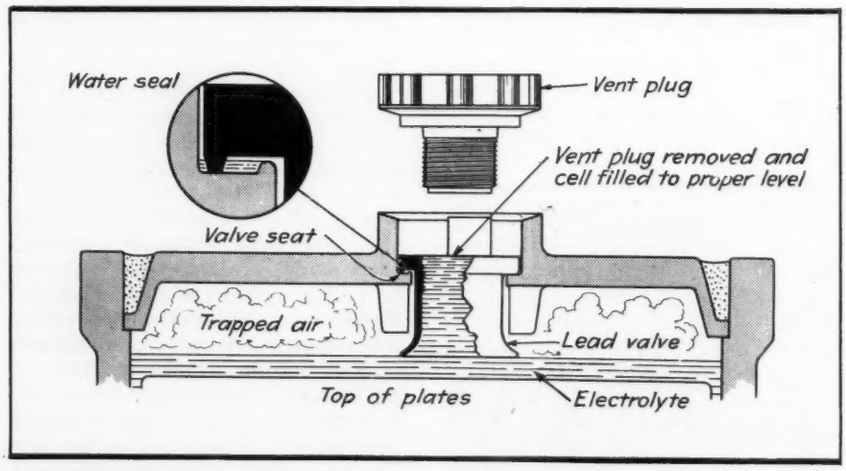
Babbitting is performed in a vacuum tunnel machine which holds 10,000 lb. of special alloy, high-lead babbitt containing 92 percent lead, kept at a precise temperature. Molten metal in the first stage, held to a definite level, serves to seal the passage and to preheat the strip. The vacuum chamber, maintained at from 6 to 7 mm. of mercury, evacuates all of the voids of the porous matrix, and enables the babbitt to reach every portion of the "spongy" structure to produce a secure mechanical and chemical attachment. At the exit stage, a mechanism maintains the thickness of the babbitt within desired limits and the molten bearing metal is "frozen" to its desired grain size by rapid chilling with jets of cold water. From the babbitting machine, the strip goes to a milling machine fitted with two milling cutters—the first a conventional down-cut, the second taking a climb cut. This operation removes sufficient babbitt to produce the desired total thickness of the strip. Sufficient stock is left to provide an average of 0.005 to 0.010 in. excess metal for final finishing. Parts are tinplated and finish-broached to a fine burnished finish.



Vacuum tunnel babbitting machine. The steel strip enters at the left, passing through a bath of 92 percent lead babbitt, then through a vacuum chamber which evacuates pores in matrix and allows full penetration of molten metal.

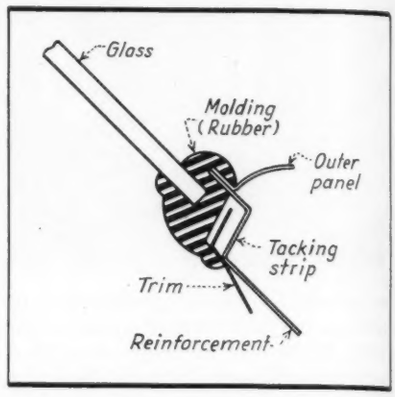


Die-cast housing is force-fitted over steel tubing in Oldsmobile steering wheel gear shift assembly. The lower edge of the die-casting is peened over a groove in the steel tube. Up-and-down movement of shifter arm is cushioned by a built-in rubber bumper. Hinge point is protected by molded rubber.

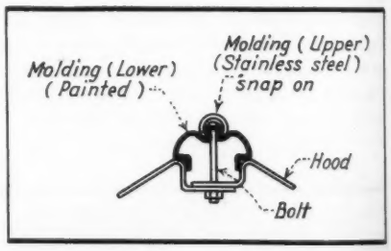


Non-overfill caps limit amount of water which can be added to the 1941 Chevrolet battery. With vent plug removed, a lead valve seats against the valve seat in the cell cover. A rim around the valve seat retains a layer of water on the seat. When the cap is removed, the lead valve drops down into the water on the valve seat and a water seal is formed. The weight of the valve on its seat, plus the water seal, traps air at the top of the cell. This prevents the addition of more than the proper amount of water. When the vent plug is screwed into place, the lead valve is raised off its seat and the trapped air and gas escape. When the vent plug is in position, a passage around the lead valve is opened which allows the charging gases to escape.

INSTRUMENT PANELS in 1941 Hudson have dials and lenses of methyl methacrylate, illuminated by light from remote sources, "piped" through the plastic. The edge-lighting properties of methyl-methacrylate have been utilized before in other fields, notably in illuminating the dial of a gasoline pump recently described in P. E. A small bulb behind the instrument panel furnishes the light to the plastic which carries it and distributes it to give a diffused and uniform illumination over the dial. The edges of the embossed word "Hudson" are roughened so that the name stands out against the plastic background. This procedure is followed in all the lettering on the dials.

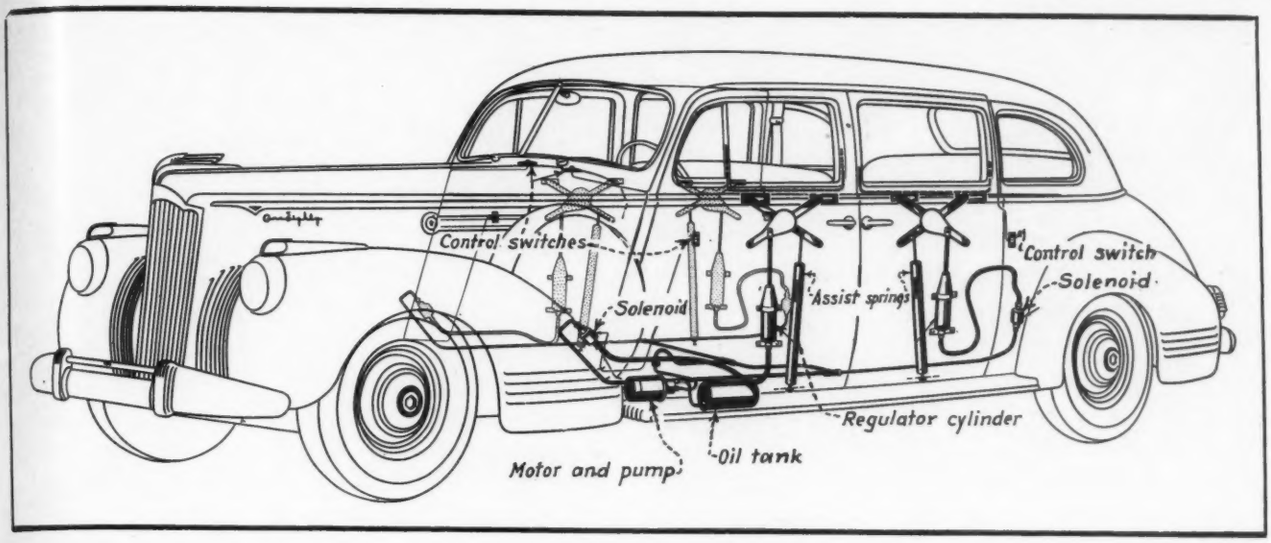


Rear window light in 1941 Studebaker. Molded rubber mounting serves as reveal, seals assembly against leakage, reduces production cost.



Center trim on Studebaker hood is chromium-plated stainless; snaps on to painted molding bolted to hood.

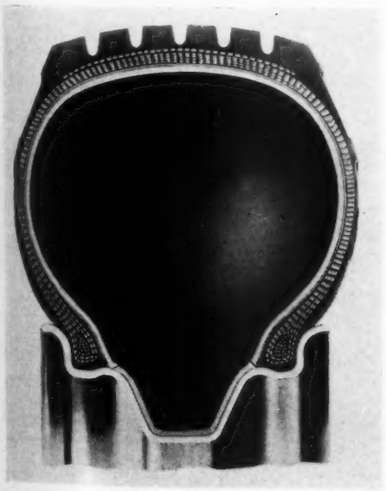
PACKARD INSTRUMENT PANEL is cellulose acetate. Speedometer and gages are covered with a panel of crystal clear plastic. Speedometer pointer is formed of an acrylic plastic with light conducting properties. An electric bulb at the hub throws light along the plastic pointer causing it to glow during the various speed ranges. Control knobs are chrome-trimmed plastic. The switch controlling the instrument panel light gives rheostatic control of the degree of illumination of the gages and clock. Steering wheel is molded hard rubber.

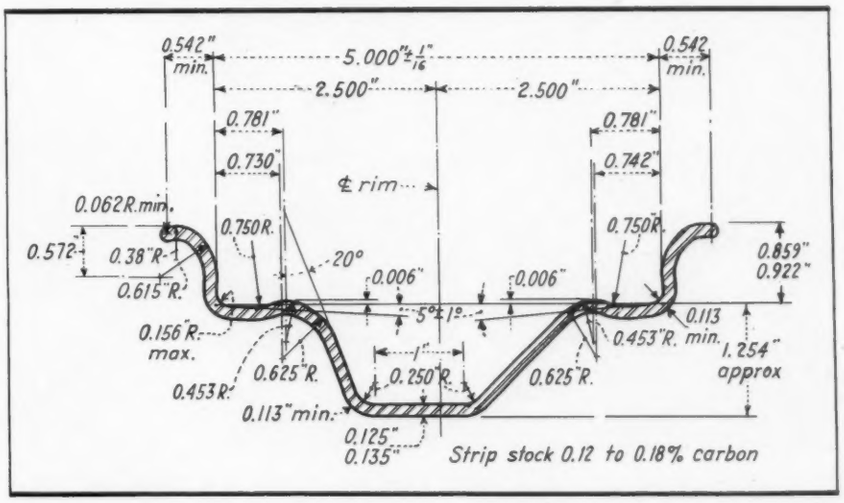


Automatic hydraulic window lift in the Packard motor and pump unit consists of a 6 volt reversible motor with a built-in direct connected gear pump which circulates hydraulic brake fluid through metallic tubing and rubber hose connections. Motor is a standard starter motor with four field coils, wound with two coils for pressure, or raising rotation, and the other two for opposite, or lowering rotation. The coils are wound to give approximately the same speed in either direction. Fluid pump is a conventional gear pump with a die-cast housing which also forms one end frame of the motor. Gears are spiral-cut for quiet operation. A non-adjustable, spring-loaded pressure regulating valve is fitted in the pump cover; a dash pot prevents chatter. This valve operates to by-pass the fluid when the window reaches its upper limit or when it strikes an obstruction. A normally closed solenoid-operated valve, fitted in the line to each window, is controlled by the operating switch of each particular door. Within each door and the division partition of limousines is a cylinder and piston assembly, connected to the conventional window-regulator mechanism, to raise or lower the window. Down stroke, which is produced by the vacuum created in the cylinder by withdrawal of the oil, is supplemented by an assist spring. Double acting control switches are provided.

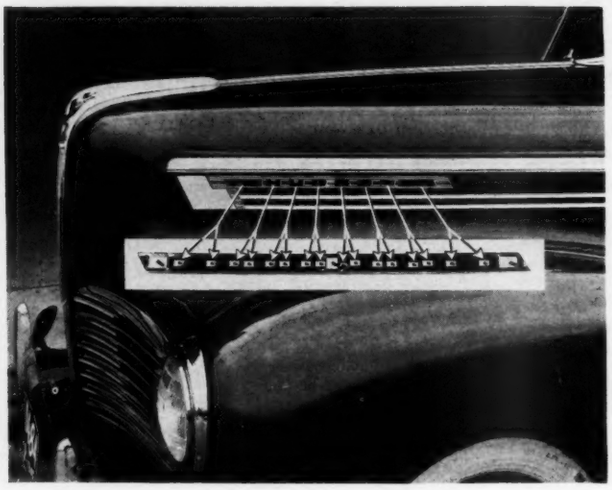
To raise any window, the proper switch is moved to the "up" position. This starts the motor, and at the same time opens the solenoid valve in the line to the window allowing oil under pressure to flow to the operating cylinder. When in a raised position, the window is held by the oil trapped in the cylinder by the normally closed solenoid valve. To lower the window, the control switch is moved to the "down" position, starting the motor in the reverse direction, again opening the solenoid valve, and withdrawing the fluid from the cylinder.

Two solenoid-operated switches are in the circuit, one for each pair of field windings, with a relay to prevent the closing of both switches at the same time. The relay is connected to give control to the wiring which lowers the window; therefore if one operating switch is held in the "down" position and at the same time some other switch is moved to the "up" position both windows will open. Two or more windows may be operated simultaneously in the same direction, but not in opposite directions, and in all cases downward movement will dominate.

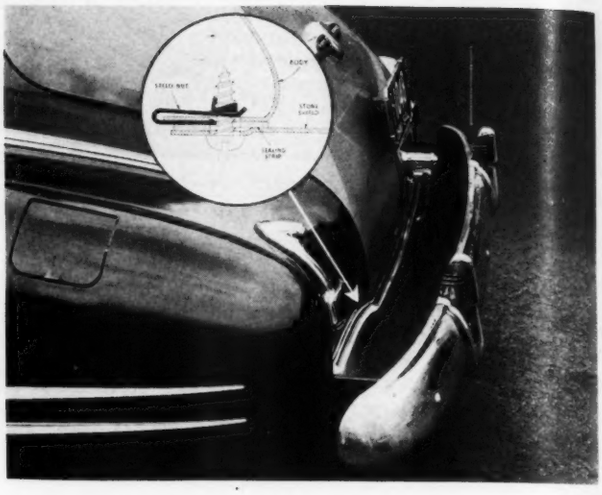




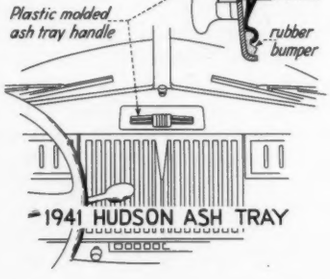
Safety rim on Chrysler Corporation cars prevents punctured tire from being thrown off, enables driver to maintain control of car in case of blowout, even at 50 to 60 mi. per hr. A small hump, formed where tire beads seat on rim, prevents the tire from sliding down into the well. On blowout, tire cannot collapse completely as beads are held by the hump and the tire rolls as if under 10 lb. air pressure. A special tool removes tire. *(Continued on next page)*

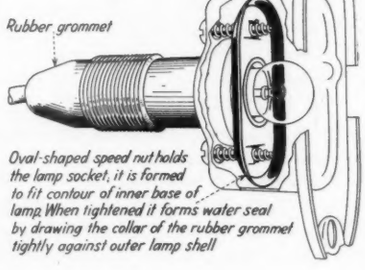


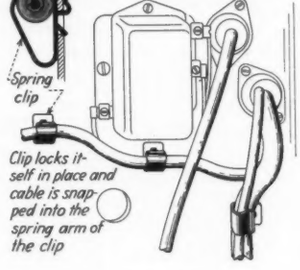
Separate letters of the name Hudson either black Tenite or die-cast are attached with integral stud and speednut.

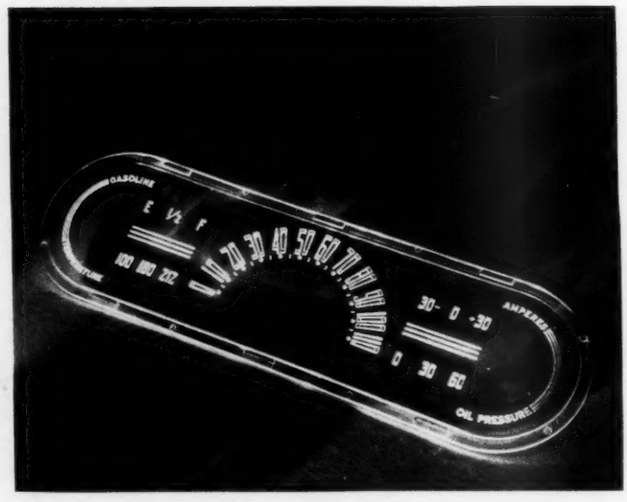


Stone shield at rear of 1941 Buick is permanently attached to body with special speednut. Assembly is vibration-proof.

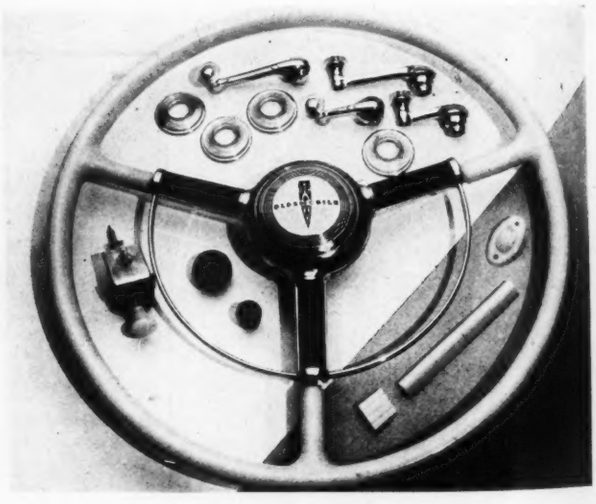

Each of 2 speed nuts holds plastic handle and rubber bumper
Integral stud of plastic handle
Plastic molded ash tray handle
rubber bumper
1941 HUDSON ASH TRAY


1941 CHEVROLET PARKING LAMP ASSEMBLY
Rubber grommet
Oval-shaped speed nut holds the lamp socket; it is formed to fit contour of inner base of lamp. When tightened it forms water seal by drawing the collar of the rubber grommet tightly against outer lamp shell


1941 CADILLAC CABLE ASSEMBLY
Cable
Spring clip
Clip locks itself in place and cable is snapped into the spring arm of the clip



Instrument-assembly lens for 1941 Oldsmobile is molded of Crystalite. Letters are edge-lighted from remote source.



Steering wheel in 1941 Oldsmobile is molded of Lumarith over steel core. Bezels are also Lumarith, handles die-cast.

« Additional mechanical features of the 1941 cars will be published in November Modern Designs section »